The Journal of the Society of Automotive Engineers

JULY, 1927

INDEX TO VOLUME XX

January-June, 1927

THE SOCIETY OF AUTOMOTIVE ENGINEERS, INC.
29 WEST THIRTY-NINTH STREET
NEW YORK CITY

Engineering
Direct

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

INDEX TO VOLUME XX, JANUARY-JUNE, 1927

	PAGES
January	1-162
February	163-322
March	323-420
April	421-528
May	529-684
June	685-826

A

INDEX TO VOLUME XX

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

Vol. XX January, 1927 No. 1

Chronicle and Comment

1927

THE officers and members of the Council extend to the members of the Society cordial greetings and best wishes for a happy year to all.

The Carnival, Bigger and Better of Course

THIS year the Carnival will be held again at Oriole Terrace, Detroit, the date being the last day of the Annual Meeting, Jan. 28. The plans for this event were revealed in the special bulletin sent on Dec. 29 by the Carnival Committee to all members of the Society.

Chairman Youngren Resigns

H. T. YOUNGREN has resigned the Chairmanship of the Buffalo Section. The Buffalo Section thus loses an able and active worker.

Mr. Youngren has accepted a position as designing engineer with the Studebaker Corporation of America, and assumed his duties on Jan. 1.

The Society Office

IT is always open house at Society Headquarters on 39th Street, but especially so during the week of the New York Automobile Show. If you have a question on standards, research, publications, meetings, or Sections, there will be someone there to discuss it with you. Also, the Engineering Library on the thirteenth floor of the building is worth a visit.

Editing Discussion of Papers

ONE of the factors causing delay in the publication of papers to be printed with the discussion is the difficulty of obtaining the approval of the members taking part in the discussion. At the Annual Meeting arrangements will be made whereby members may read and approve the stenotype record of the discussion within 2 hr. after each session. The stenotype reports of the sessions will be held at the S.A.E. Information Desk.

Discussion At Annual Meeting

ADVANCE copies of most of the papers to be presented at the Annual Meeting will be available in mimeographed form so that those who desire to discuss the various subjects may review the papers before the meeting. Copies are being sent to all members interested.

The Meetings Committee keenly appreciates the need of written discussion submitted in advance of the meeting and trusts that all members interested in the subjects to be presented at the Annual Meeting will obtain advance copies of the papers and submit written discussion.

The Annual Meeting

THE complete program for the Annual Meeting is given on p. 45 of this issue. Including the names of Alan Fenn, Charles E. Weymann, F. Sergardi, Johannes Plum, and G. Constantinesco, the program has a distinctly European flavor. The Constantinesco transmission paper to be presented by R. K. Jack, has been prepared by him with the approval of Mr. Constantinesco.

The program of Sessions arranged by the Meetings Committee seems to have particular appeal to the members interested in the various phases of automotive engineering. It is, therefore, difficult to single out any of the sessions for special mention.

R. H. Grant To Speak at Annual Dinner

R. H. GRANT, vice-president of the Chevrolet Motor Co., will make the principal address at the Annual Dinner at the Hotel Astor on Thursday, Jan. 13. His subject is What Sells Motor-Cars. When Mr. Grant was with the Dayton Engineering Laboratories Co., the members had the privilege of hearing him, in 1922, at the Chicago Service-Meeting Dinner. In view of Mr. Grant's new connection and the record of the Chevrolet Company during the last year, the Dinner Committee is to be congratulated on prevailing upon Mr. Grant to address the members again. The complete program for the 1927 dinner is given on p. 44 of this issue.

Sections Committee

THE first meeting of the 1927 Sections Committee will be held on Wednesday, Jan. 26, at the General Motors Building, Detroit. The representatives appointed are V. G. Apple, Dayton Section; George T. Briggs, Indiana Section; F. K. Glynn, Metropolitan Section; C. O. Guernsey, Pennsylvania Section; A. W. Herrington, Washington Section; P. B. Jackson, Buffalo Section; E. B. Moore, Southern California Section; Walter S. Nathan,

Milwaukee Section; E. V. Rippingille, Detroit Section; E. W. Weaver, Cleveland Section; Glenn S. Whitham, New England Section; Robert E. Wilson, Chicago Section; and Edwin C. Wood, Northern California Section

Reservations

JANUARY, bringing the Annual Dinner and Meeting, the Metropolitan-Section Dinner-Meeting, the Carnival, and the railroad and hotel reservations involved, might well be called the month of reservations.

The number of reservations received for the Annual Dinner at the time of going to press is over 900. Without doubt the capacity of the Astor ballroom will be taxed to the limit, even though arrangements have been made to use the recently remodeled balcony.

Almost a month before the event, over 700 reservations have been received for the Carnival. Every seat will be sold and more demanded as in past years. Reservation blanks for the Dinner and the Carnival were sent to the members with the *Meetings Bulletins* of Dec. 8 and 29.

Two Sections for This Number

THIS issue of THE JOURNAL consists of two sections, Section 1 being the regular monthly issue for January and Section 2 the Index to Vol. XIX. From December, 1917, when the Index to Vol. I was printed, until June, 1926, the index to each half-yearly volume of THE JOURNAL has appeared in the last issue of the volume, June and December, respectively. Six months ago a departure from this practice was made to facilitate the mailing of the June, 1926, issue containing the news account of the Semi-Annual Meeting at French Lick Springs, Ind., held on the first 4 days of the month. This change was so successful, enabling that issue to be mailed in less than a week after the close of the meeting, that a similar procedure was followed in connection with the December, 1926, issue of THE JOURNAL containing the news account of the Transportation and Service Meeting.

Any member who fails to receive a copy of the semi-annual index as part of this issue of THE JOURNAL is requested to notify the Publication Department at the Society's headquarters in New York City so that a copy can be mailed to him.

Do You Wish To Receive TRANSACTIONS Regularly?

SOME of the members have inquired if, instead of ordering each Part of TRANSACTIONS when the time for placing orders arrives, an arrangement cannot be made whereby one order can be placed to cover all future Parts. To meet the wishes of these members, the order blank for Part II of Vol. 20 (1925), which will be mailed to the members in the near future, will contain space for a request to be placed on a list to receive Parts of TRANSACTIONS as they are issued from time to time in the future, payment at the rate of $2 per Part to be made upon presentation of a bill.

Part II of Vol. 20 of TRANSACTIONS will consist of approximately 928 pp. and will contain papers presented at the 1925 Semi-Annual, Aeronautic, Production, Automotive Transportation, and Service Meetings of the Society; other papers presented at Section Meetings; and contributed articles that were published in THE JOURNAL from July to December, 1925, inclusive; comprising a total of 43 papers and articles, including discussion. Volumes of the TRANSACTIONS are now sent only to members ordering them at the nominal price of $2 per part. To be assured of receiving a copy of this Part, members should place their orders by March 12.

Standards Committee Meeting

THE Standards Committee 1927 Annual Meeting will convene on the fifteenth floor in the west wing of the General Motors Building, Detroit, on Tuesday morning, Jan. 25. It will be called to order promptly at 9:30 to allow as much time as possible to dispose of the reports of the several Divisions before the luncheon hour. The regular technical discussions of the Society commence on Tuesday afternoon and it is desirable that the Standards Committee session be completed before then. The Divisions of the Standards Committee that have submitted reports are:

Aeronautic
Axle and Wheels
Electrical Equipment
Engine
Iron and Steel
Lighting
Motor-Truck
Parts and Fittings
Production
Screw-Threads
Transmission

All members of these and other Divisions of the Standards Committee are urged to be present at the opening hour; and others who are directly interested in the Divisions' reports included elsewhere in this issue of THE JOURNAL will be welcome.

Abrasives in Oil and Wear

FOREIGN material in crankcase oil has always been recognized as a source of injurious wear in bearings and working parts of engines. The extent of this recognition, as well as the extent of damage caused, is best indicated by the amounts of attention and money that have been devoted in recent years to developing various devices for filtering or otherwise purifying engine lubricants during operation. The problem, for the most part, has been intimately associated with the question of crankcase-oil dilution and, while various devices and methods have been introduced to combat the two phenomena jointly and separately, very little has been done in the direction of actually measuring the effects of abrasives in oil and the little that has been accomplished has been largely connected with road runs in performance tests or, more frequently, with vehicles in regular service.

S. A. McKee's report, contributed by the Bureau of Standards, and presented in this issue in the Automotive Research section, on the subject of Performance Characteristics of Journal-Bearings with an Abrasive in the Lubricant, is particularly timely as indicating by scientifically conducted laboratory tests the variations in the coefficient of friction produced by the introduction of abrasive material in the oil and also the influence of the viscosity of the oil.

While the investigation has not been sufficiently extensive to provide conclusive data covering a wide range of conditions, the results indicate what can be expected from further research in this very important field.

New Motorcoach Regulations in New Jersey

A COMMITTEE of the New Jersey Board of Public Utility Commissioners, which drafted a revision of the New Jersey Motorcoach Regulations, submitted the draft to the Society for comment at the time of the

(*Concluded on p. 66*)

AUTOMOTIVE RESEARCH

The Society's activities as well as research matters of general interest are presented in this section

EFFECT OF ABRASIVE IN LUBRICANT

Bureau of Standards Conducts Tests to Determine Performance Characteristics

We are indebted to the Bureau of Standards for the following contribution by S. A. McKee covering the results of experiments undertaken at the Bureau. The general scheme of conducting the tests involves acceptance of the common theory of lubrication of complete journal-bearings and actual operation of bearings under conditions simulating practical applications, with oils of different viscosities and with and without the addition of an abrasive.

The results of these tests will, it is hoped, stimulate further research on the interrelated effects of viscosity and contamination, foreign matter and other influences, leading to a fuller and better understanding and correct appraisal of the fundamental factors affecting journal wear.

REPORT BY S. A. MCKEE[1]

A study of the performance characteristics of journal-bearings when an abrasive is in the lubricant was undertaken to obtain an indication as to what effect an increase in the viscosity of the lubricant might have on the performance of the bearings of automobile engines when diluted or low-viscosity crankcase-oil is contaminated with road dust or other solid matter.

The accepted theory[2] of lubrication of a complete journal-bearing free from end leakage shows that, in a typical bearing operating under steady conditions such that a complete film of the lubricant entirely separates the journal or shaft from the bearing, the shaft will take up some such position as is shown by the exaggerated cross-section in Fig. 1. The distance X between the shaft and the bearing at the point of nearest approach, is dependent upon $\mu n/p$, where $\mu =$ the absolute viscosity of the lubricant in the bearing, $n =$ the speed of the shaft, and $p =$ the pressure on the projected area of the bearing. With a given bearing and shaft, X increases with an increase in the speed of the shaft, decreases with an increase in the load, and increases with an increase in the absolute viscosity of the lubricant. Thus, when operating under conditions of load, speed and viscosity such that the distance X is less than the greatest dimension of the particles of abrasive carried along in the oil stream, the bearing performance might be expected to be of a different nature than when operating with the same bearing and shaft under the same load and speed but using an oil with a viscosity great enough to increase X until it becomes greater than the greatest dimension of the largest particle of abrasive. In the latter case it would be expected that the performance characteristics of the bearing would be nearly the same as if a clean oil of the same viscosity were used, for the effect of the particles striking the surfaces of the shaft and bearing would probably be practically negligible. In the former case, however, a marked difference would be expected because the oil stream would have a tendency to force or wedge the particles through the construction at X, with a corresponding increase in the frictional loss in the bearing.

A convenient method for comparing the performance of journal-bearings is to obtain the f or coefficient of friction versus $\mu n/p$ curves of the bearings by a journal-bearing friction-machine. The f-versus-$\mu n/p$ curve is based on the theory that for any given journal and bearing, f also is a function of $\mu n/p$ so long as operating conditions are not severe enough to rupture the oil film partially or completely.

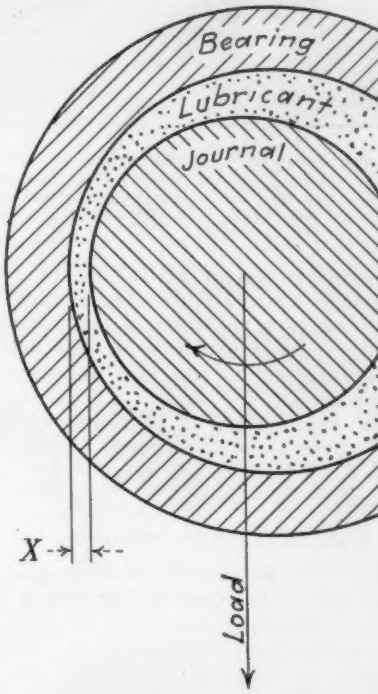


FIG. 1—DIAGRAMMATIC REPRESENTATION OF TYPICAL BEARING

The Distance X Measures the Point of Nearest Approach of the Shaft to the Bearing with Complete Film of Lubricant in Complete Journal-Bearing Under Steady Conditions

This method is especially adaptable to the problem in hand where both f and X are determined by the value of $\mu n/p$ for an oil free from abrasives. Thus the difference between the curve when using clear oil and that when using the same oil plus abrasive, is for any given value of $\mu n/p$ a measure of the added friction caused by the presence of the abrasive.

It should be noted that the quantity $\mu n/p$ is dimensionless and its numerical value for any specific condition of operation is independent of the system of units employed, provided that the system is consistent. However, as a matter of convenience, the quantity ZN/p, suggested by Wilson and Barnard, has been substituted in the remainder of this discussion. This quantity, ZN/p, is specifically defined by the common engineering units,

$N =$ the speed of the journal or shaft, in revolutions per minute
$p =$ the pressure on the projected area of the bearing, in pounds per square inch
$Z =$ the viscosity of the lubricant in the bearing, in centipoises

Since, in this term a consistent system of units is not employed, its numerical value for any specific condition of operation is not the same as the theoretical modulus $\mu n/p$. It is, however, dimensionless, and directly proportional to $\mu n/p$, the factor of conversion being approximately

[1] Published by permission of the Director of the Bureau of Standards.

[2] See *Zeitschrift fur Mathematik und Physik*, vol. 50, p. 97; *Transactions of the American Society of Mechanical Engineers*, vol. 37, p. 167; and THE JOURNAL, July, 1922, p. 49.

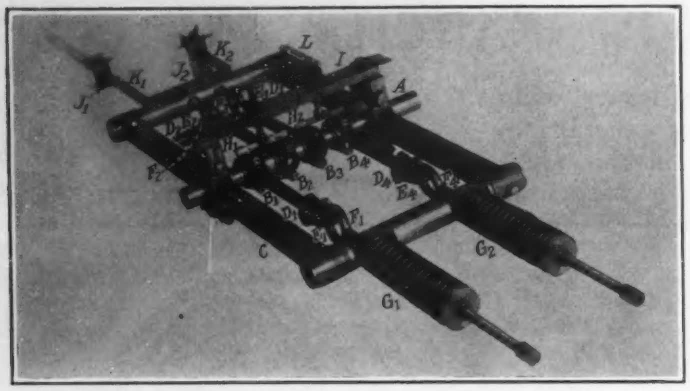


FIG. 2—MOUNTING OF TEST BEARINGS
Connecting-Rod Crankpin Bearings B_1, B_2, B_3, and B_4 Are Mounted on Shaft A, with Load Applied by Compressing the Calibrated Coil-Springs G_1 and G_2. Frictional Torque Is Measured by Counterpoises J_1 and J_2

151×10^{-10}. This conversion factor of 151×10^{-10} is the product of the two factors, 2π and approximately 24×10^{-10}. The factor 24×10^{-10} converts centipoises to pounds per square inch divided by radians per minute, this being the viscosity unit that corresponds to the unit of pressure of the pound per square inch. The factor 2π converts the number of revolutions per minute to radians per minute, this being the unit of speed that corresponds to the pounds per square inch divided by radians per minute unit of viscosity. Thus $\mu n/p = 151 \times 10^{-10}\, ZN/p$.

TESTING EQUIPMENT AND METHOD

A photograph of the friction machine used in this work is shown in Fig. 2 and one of the complete set-ups, in Fig. 3.

The heat-treated steel test-shaft A, which is made of a high-carbon-tungsten tool-steel, oil-quenched at 1550 deg. fahr., tempered 30 min. at 900 deg. fahr., and having a Brinell hardness of about 350, is mounted in a lathe driven by a variable-speed direct-current motor. One end of the shaft is centered in a four-jaw chuck on the headstock and the other is supported by a bronze bearing set in a steady-rest. Temperature control of the shaft is provided by making it hollow for a considerable portion of its length to allow for water circulation when necessary.

The test bearings, four in number, B_1, B_2, B_3, and B_4, Fig. 2, are the babbitt-metal-lined crankpin bearings of automobile engine connecting-rods with a nominal bore of 1¼ in. and length of 1¼ in. The approximate composition of the babbitt is 85.0 per cent of tin, 7.5 per cent of copper and 7.5 per cent of antimony. The bearings are of stock finish except that oil-grooves are omitted and lie along the shaft

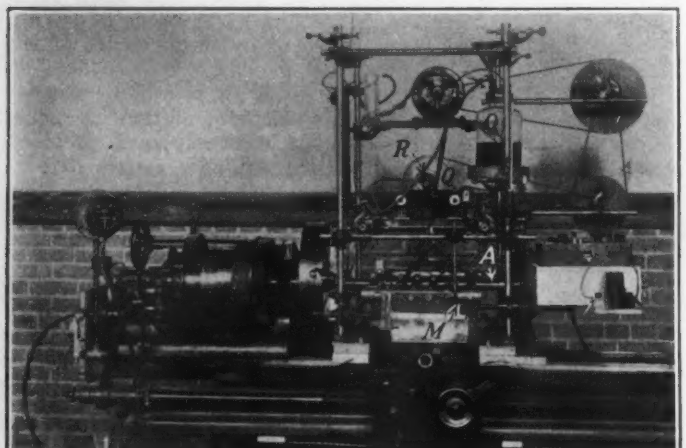


FIG. 3—COMPLETE TESTING EQUIPMENT
Clear Oil Is Fed by Air Pressure through Tubes from Glass Bottle O and Oil Containing Abrasive Material from Slowly Rotating Bottle Q

on 2-in. centers with the center lines of the connecting-rods in a horizontal plane. The wristpin ends of the rods are fastened to a special frame, C, by pins D_1, D_2, D_3, and D_4; clevises E_1, E_2, E_3 and E_4; and eyebolts F_1, F_2, F_3, and F_4 in such a manner that rods B_1 and B_4 lie on the opposite side of the shaft from rods B_2 and B_3.

Load is applied to the bearings by compressing the calibrated coil-springs G_1 and G_2 mounted on the sliding eyebolts F_1 and F_4, fitted to rods B_1 and B_4. The reaction of these springs is taken up by rods B_2 and B_3, and transferred to the frame by the clevises E_2 and E_3 and fixed eyebolts F_2 and F_3. This method provides equal loading on all bearings since they are equally spaced. The weight of the frame is transferred to the bearings by the equalizers H_1 and H_2 mounted on the cross-beam I fastened to the frame just above the shaft.

When the shaft is in motion and the load applied, the bearings and frame act as a rigid unit which is allowed to float on the shaft and tends to rotate around the axis of the shaft due to the frictional torque set up in the bearings. This frictional torque is measured by the counterpoises J_1 and J_2 mounted on the graduated beams K_1 and K_2, the spirit-level L giving an indication of when the system is in balance, and the stop M, Fig. 3, facilitating the balancing process. A feature of this design is that with the bearings and frame floating freely on the shaft there is not the possibility of error in measuring the frictional torque that there is in machines where the motion of linkages is necessary for this determination.

The working temperature of the bearings is measured by the copper-constantan thermocouples soldered to the loaded side of bearings B_1 and B_3, Fig. 2.

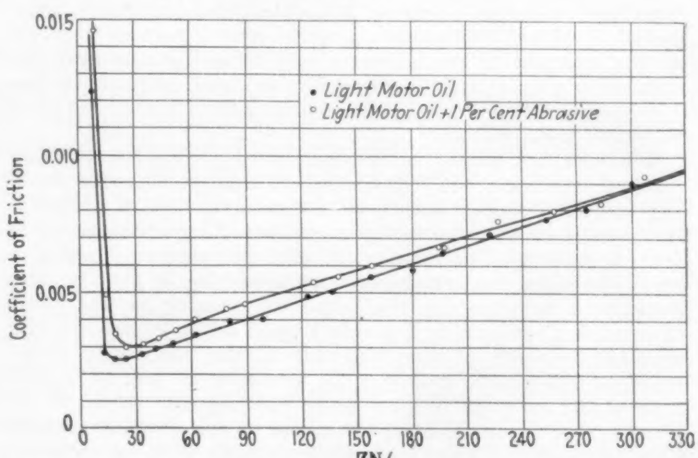


FIG. 4—COEFFICIENT OF FRICTION PLOTTED AGAINST THE ZN/p VALUE OF A LIGHT MOTOR-OIL
One Curve Was Obtained with Clear Oil and the Other with the Same Oil after 1 Per Cent of Abrasive Was Added

Oil is fed to the bearings through tubes placed in the center of their unloaded sides. Branch lines of rubber tubing lead to a 1-gal. glass bottle, O, Fig. 3, containing the clear oil, and to a 2-gal. glass bottle, Q, containing the oil mixed with abrasive material. A means for keeping the abrasive suspended in the oil is provided by mounting the bottle in a horizontal position on cork-tired rollers and slowly rotating it by a belt drive from a small direct-current motor. The oil is forced out of these bottles by the house air-pressure of about 7 cm. (2.76 in.) of mercury, a special bearing, R, being inserted in the stopper of the rotating bottle to provide for the admittance of the air-pressure and oil-feed lines, which are stationary.

The abrasive material chosen for use in these tests is a form of diatomaceous earth, a grayish white powder consisting mostly of silica. The greatest dimensions of the particles range from 0.00004 to 0.00010 in. Of the materials available, this offered the most promise of approaching the probable degree of hardness and fineness of the solid matter suspended in a used crankcase-oil. The concentration chosen was 1 part, by weight, of abrasive to 100 parts of oil. This

concentration is considerably in excess of those usually found in used crankcase-oils, but it was deemed advisable to use this high concentration to make sure that the effect of the abrasive could be noted.

In making a test run, the clear oil was fed to the bearings and the machine was brought to steady conditions under a predetermined load and speed to give the desired value of ZN/p. A determination of the frictional torque was then made. This was done by taking observations of the positions of the counterpoises necessary to bring the frame to a balance in a horizontal plane, with the shaft rotating first in one direction and then the other, the difference between the average of six readings taken when the shaft was rotating in one direction and the average of six readings taken

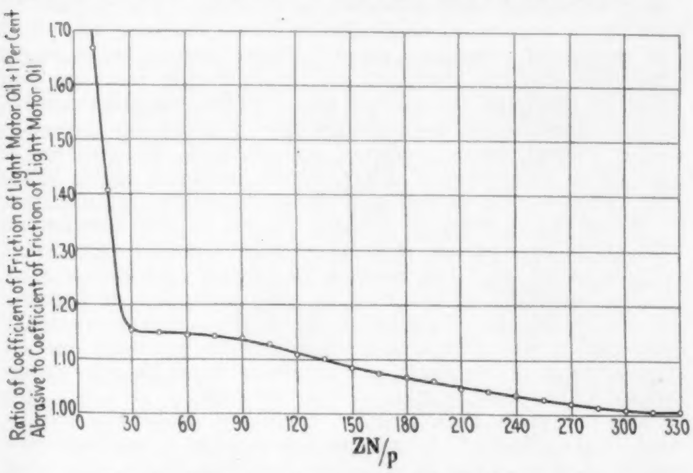


FIG. 5—EFFECT OF ABRASIVE IN LIGHT MOTOR-OIL
The Ratio of the Coefficient of Friction When Using the Oil Mixed with Abrasive to the Coefficient of Friction When Using Clear Oil Is Plotted against ZN/p

when the rotation of the shaft was reversed being twice the average frictional torque of the four bearings. A check was then made by taking 6 more readings with the shaft rotating in the original direction and these 6 check-readings were used with the 12 prior ones in obtaining the final average value for the frictional torque. To prevent injury to the shaft or bearing when reversing, the driving motor was "plugged in" in the opposite direction before the test shaft had come to rest. To eliminate any error due to the possible dragging effect of the oil-lines, the six readings of the positions of the counterpoises necessary to bring the frame to balance with the shaft rotating in a given direction were taken with the frame tipped alternately in opposite directions. When this determination of the frictional torque had been made, the flow of the clear oil was stopped, the oil-lines were drained, and the oil mixed with abrasive was fed to the bearings, this being done without stopping the machine. After allowing a few minutes to be sure that the gritty oil had entirely displaced the clear oil in the bearing, a determination of the frictional torque was made in the same manner as before.

Tests with Light Motor-Oil

In the first series of test runs the performance of a light motor-oil having a viscosity of about 200 sec. Saybolt universal at 100 deg. fahr. was compared to that of the same oil plus the abrasive. The runs were made in the order of the decreasing value of ZN/p. The f-versus-ZN/p curves of the bearings operating with these lubricants are shown in Fig. 4. It will be noted that the curve obtained when using the clear oil has characteristics common to curves obtained with this type of bearing. The points lie in a straight line at the right of the curve in the region of "stable lubrication" where a complete "fluid film" of the lubricant separates the shaft from the bearing. As ZN/p decreases, f decreases until the minimum point is reached, to the left of which f rises rapidly in the region of "unstable lubrication" where the fluid film of the lubricant has broken down. On the other hand, the curve obtained when using oil mixed

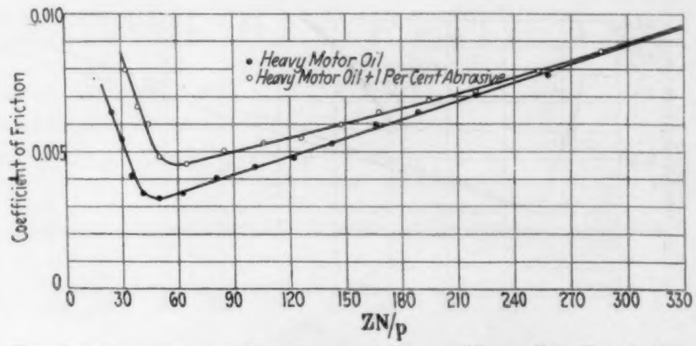


FIG. 6—COEFFICIENT OF FRICTION OF A HEAVY MOTOR-OIL, CLEAR AND WITH 1-PER CENT ABRASIVE, PLOTTED AGAINST ZN/p
The Effect of Abrasive Is Practically Negligible at the Higher Values of ZN/p and This Effect Increases as ZN/p Decreases Until It Assumes Rather Large Proportions as the Conditions of Unstable Lubrication Are Approached

with abrasive is different. At the extreme right it lies very close to the curve for the clear oil, but as ZN/p decreases it gradually slopes away from the clear-oil curve and the point of minimum value of f occurs at a higher value of ZN/p.

Due to the particular form of these curves, the effect of the abrasive in increasing the coefficient of friction is not clearly shown. For example, where ZN/p equals 30 the distance between the two curves is about the same as the distance between them where ZN/p equals 210, yet at 30 this difference represents an increase in friction of about 15 per cent, while at 210 it means an increase of but 5 per cent. The effect of the abrasive is more clearly shown in the curve in Fig. 5 where R or the ratio of the coefficient of friction when using the oil mixed with abrasive to the coefficient of friction when using clear oil is plotted against ZN/p.

In a second series of test runs, a dierent set of bearings was used and the effect of the abrasive was noted when using a heavy motor oil having a viscosity of about 385 sec. Saybolt Universal at 100 deg. fahr. The f-versus-ZN/p and R-versus-ZN/p curves obtained under these conditions are shown in Figs. 6 and 7. These curves are very similar to thsoe obtained from the first series of runs. At the higher values of ZN/p the effect of the abrasive is practically negligible and this effect increases as ZN/p decreases until it assumes rather large proportions as the conditions of unstable lubrication are approached.

There is some experimental evidence that some of the properties of the oil containing abrasive deviate to a very slight extent from those of a true viscous liquid, and if this were not taken into account the dierence in the f-versus-ZN/p curves of the clear oil and the oil plus abrasive could be attributed to the difference in "fluid-friction" of the two lubricants. In computing the values of ZN/p for operation

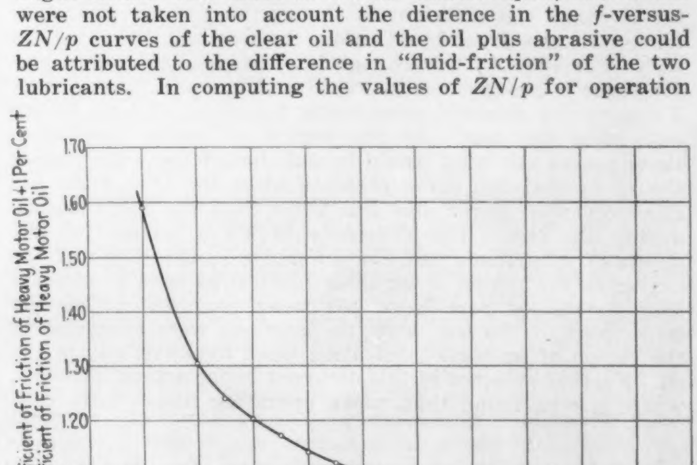


FIG. 7—EFFECT OF ABRASIVE IN HEAVY MOTOR-OIL
The Differences Shown by This Curve and by Fig. 5 Were Due to the Particles of Abrasive Rubbing against the Metallic Surfaces Rather than to the Internal Friction of the Fluid Itself

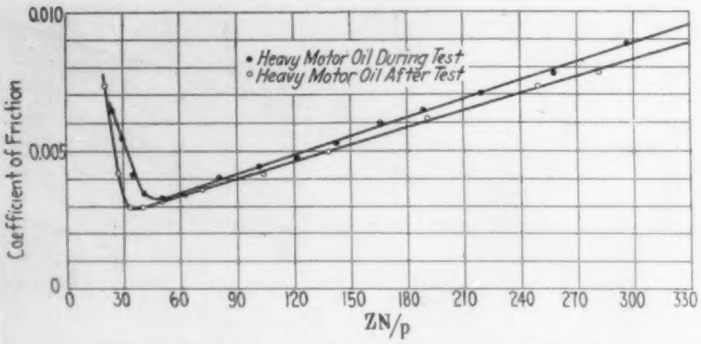


FIG. 8—HEAVY MOTOR-OIL DURING AND AFTER TEST
These Curves Tend To Show That the Most of the Wear Occurred in the Regions of Unstable Rather than Stable Lubrication, and It Is This Region of Stable Lubrication That Is Most Important in This Particular Problem

with the oil plus abrasive, however, a correction for this difference was made by using the value of the "apparent viscosity" at the temperature of operation. This correction, although small, is ample, for the value of the apparent viscosity was obtained when the liquid was undergoing an average rate of shear considerably less than the average rates of shear it experienced in the bearing, and experimental evidence indicates that the percentage difference in fluid friction between the oil plus abrasive and the clear oil becomes less when the rate of shear is increased. Thus, the differences shown by the curves in Figs. 5 and 7 were due to the particles of abrasive rubbing against the metallic surfaces rather than to the internal friction of the fluid itself.

PERFORMANCE CHARACTERISTICS OF JOURNAL-BEARING WITH AN ABRASIVE IN THE LUBRICANT

When the tests were started, it was believed that because of the short duration of a run, the wear incurred during a series of runs would be too small to be measured but, after the first series was completed, it was found that the performance characteristics of the bearings had been changed considerably. Therefore, just as a matter of interest, an attempt was made to measure the wear that occurred during the second series of runs. Sufficient measurements were taken as accurately as possible on both the bearings and shaft to determine the average clearance both before and after the series of runs and it was found that, while the wear on the shaft was practically negligible, the wear on the bearings was sufficient to change the average clearance-diameter value from 1/663 before the test to 1/597 after the test. The effect of this change in clearance on bearing performance is brought out in Fig. 8 which shows the f-versus-ZN/p curves obtained when using the clear oil both during and after the test. In the region of stable lubrication these curves are what would be anticipated from lubrication theory in that the curve obtained after the test, when the clearance was larger, has less slope than the one obtained during the test. The difference in ZN/p values for the points of minimum f and the difference in the slopes of the curves in the region of unstable lubrication tend to indicate that the major portion of the wear occurred during the latter part of the test when the bearings were operating in the region of unstable lubrication when abrasive was in the oil. Further evidence of this occurred in the actual test-runs, where it was found that, when operating under these conditions, the frictional torque was slowly but continually decreasing and the observations for measuring the torque had to be taken "on-the-fly," as it were.

It might be well to point out that the existence of a relation between f and ZN/p in the region of stable or complete-film lubrication, depends on the assumption that the physical conditions of the shaft and bearing remain constant, and hence that the f-versus-ZN/p curves shown in Figs. 4 and 6 are not what might be called "true" curves, since probably each point on any one curve has been obtained under slightly different conditions of clearance. However, the curves in Fig. 8 tend to show that the most of the wear occurred in the region of unstable rather than stable lubrication, and it is this region of stable lubrication which is most important in this particular problem. The method by which the runs were made also tends to minimize this effect of change of clearance, since a direct comparison between the clear and the gritty oil was made during the same run without changing the load or speed. Thus the difference between these curves, even though they are not true f-versus-ZN/p curves, should indicate the effect of the abrasive in the oil at any value of ZN/p in the region of stable lubrication practically as well as if the clearance had not changed.

While the increase in friction caused by the abrasive in the oil is in itself of interest, a point possibly of greater significance is that this increase in friction furnishes a qualitative indication of the relative wear occurring during different conditions of operation, since it is evident that the increase is caused by actual rubbing of the particles against the surfaces of the shaft and bearing, rather than to an increase in the fluid friction of the lubricant.

The curves in Figs. 5 and 7 furnish a qualitative indication of the effect of viscosity on the wear in a bearing when an abrasive is in the oil and show that it is possible to build up an oil-film in a bearing to a thickness sufficient to practically nullify the wearing effect of a fine abrasive present in the lubricant in a quantity considerably in excess of the amounts usually found in used crankcase-oils. They agree fairly well in that this thickness of film apparently is built up at approximately the same value of ZN/p in the two sets of bearings used in these tests. This value of ZN/p is, of course, typical only of the particular conditions of length, diameter, clearance, and the like that are present in these bearings, and of the hardness, size and amount of abrasive material in the lubricant, and it is to be expected that wide variations both above and below this value would be found under other conditions. It is very probable, too, that in many cases this value would be so high that it would be impracticable to use an oil with a viscosity high enough to keep ZN/p at all operating conditions above this critical value. However, even though this should be true in every case, the curves also show that the wear on the bearing per revolution of the shaft tends to decrease as the value of ZN/p increases.

PROMISING FIELD FOR FURTHER RESEARCH

Due to the fact that these tests do not cover the complete range of operating conditions present in actual engines and that it is practically impossible to determine what all of these conditions may be, it would not be justifiable to predict that an appreciable increase in the length of life of an engine would be effected by the use of the heaviest oil that its lubricating system could successfully handle. These results do indicate, however, that there is at least a promising field for further research with reference to the factors affecting the wear of journal-bearings.

STANDARDS COMMITTEE DIVISION REPORTS

The following Division Reports will be submitted to the Standards Committee for approval at the Annual Meeting

STANDARDS COMMITTEE MEETING JAN. 25

To Be Held at Detroit during the Annual Meeting of the Society

In this issue of THE JOURNAL are printed reports that have been prepared for submission to the Standards Committee and the Society by 11 Divisions of the Standards Committee since the Semi-Annual Meeting of the Society last June. These include reports that have been submitted to the Society by Sectional Committees functioning under the procedure of the American Engineering Standards Committee, for which the Society is a sponsor. The Society's Standards Committee Regulations require that such reports be assigned to Divisions of the Standards Committee and that the same procedure be followed throughout as for reports originating in the Divisions. In some cases, these reports are submitted also for approval and adoption by the Society as part of the S.A.E. Standards as well as for American Standards under the procedure of the American Engineering Standards Committee.

All of the reports are submitted at this time for approval after having been thoroughly considered by their respective Divisions and as wide publicity as possible given them by publication in this or previous issues of THE JOURNAL. The reports as now presented are believed to be in acceptable form and any proposed changes should be only in the nature of important constructive ones and carefully considered. Under the Standards Committee procedure, these reports may be approved as presented, amended within limitations or referred back to their respective Divisions for sufficient reason. The action taken on them by the Standards Committee will be passed upon by the Council and the general business session of the Society, looking toward their approval for submission to letter ballot of the members of the Society as the final step in the procedure. The letter ballot will be counted 30 days following the Standards Committee Meeting and if affirmative, the reports will then be published in the S.A.E. HANDBOOK.

Rejection or major changes in any of the reports will require that they be sent back to their Divisions and that they cannot be passed upon before the Semi-Annual Meeting of the Society next summer. In voting on the reports the Standards Committee Regulations require that only members of the Standards Committee do so.

BRAKE NOMENCLATURE EXTENDED

Axle and Wheels Division Proposes Nomenclature Covering Four-Wheel Brakes

Some time ago the suggestion was made that the present S.A.E. Automobile Nomenclature for Wheels, p. K18 of the S.A.E. HANDBOOK, be extended to include reference to four-wheel brake-drums inasmuch as four-wheel brakes had become very extensively used on automobiles. The suggested addition to the nomenclature was referred to the Division members and has been approved by them with the recommendation that it be incorporated in the present standard as follows:

> On p. K18 of the S.A.E. HANDBOOK, under Division XV, Group 1, following the last item, include "Front-wheel brake-drums (if front-wheel brakes are used)".
>
> Under Group 2 for rear wheels, change the term "Wheel brake-drums" to read "Rear-wheel brake-drum".

AXLE AND WHEELS DIVISION PERSONNEL

C. C. Carlton, *Chairman*	Motor Wheel Corporation
L. R. Buckendale, *Vice-Chairman*	Timken-Detroit Axle Co.
C. E. Bonnett	Tire & Rim Association of America
G. W. Carlson	Eaton Axle & Spring Co.
G. W. Harper	Columbia Axle Co.
E. R. Jacobi	Hayes Wheel Co.
O. A. Parker	Parker Wheel Co.
W. F. Rockwell	Wisconsin Parts Co.
George Walther	Dayton Steel Foundry Co.

AERONAUTIC STANDARDS CANCELLED

Work to Be Limited to Specifications That Promote Interchangeability

At the Aeronautic Division meeting held on Sept. 1, 1926, in Philadelphia, it was decided that the policy of the Aeronautic Divisioh should be to limit aeronautic standardization to specifications that promote interchangeability. The number of materials used in aircraft construction is so large that to cover them completely at this time would require an amount of effort that would be out of proportion to the value of the specifications.

With reference to the material specifications adopted by the Society during 1917 and 1918 and still retained in the S.A.E. HANDBOOK, it was felt that in line with the policy decided upon these specifications should be cancelled. The specifications involved and the pages on which they appear in the S.A.E. HANDBOOK follow:

Specification	Page
Rubber Hose for Gasoline	C48
Steel Wire Cable	C65
Reels for Cable	C69
Round High-Strength Steel Wire	C70
Steel Cable Loops	C74
Cellulose Acetate Dope	D131
Cellulose Nitrate Dope	D134
Spar Varnish	D136

Inasmuch as the conferences held periodically by the Army and Navy Air Services are resulting in unifying the Army and Navy Specifications, it was recognized that the Division should undertake no standardization work that would conflict in any way with the work of these conferences. As the specifications adopted by the Army-Navy Conferences will not cover powerplants of less than 200 hp., it was understood that the Aeronautic Division would have to extend many of the AN Standards adopted to meet the requirements of commercial aircraft equipped with the smaller types of engine. That proper consideration might be given to determining the AN Standards that should be adopted by the Society as S.A.E. Recommended Practice and to the form in which they should be adopted, it was considered desirable to appoint a Subdivision to review all the AN Standards and to recommend to the Division such standards as should be adopted as S.A.E. Recommended Practice, as well as the form in which they should be adopted.

The Aeronautic Division, the personnel of which is printed

below, has therefore recommended that the present aeronautic specifications listed above be cancelled.

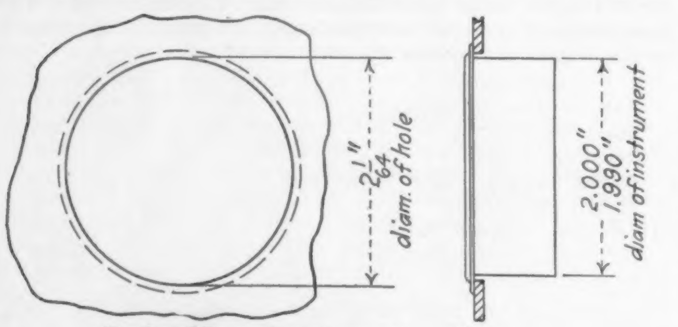
AERONAUTIC DIVISION PERSONNEL

E. P. Warner, *Chairman*	Assistant Secretary of the Navy in Charge of Aviation
V. E. Clark, *Vice-Chairman*	Consolidated Aircraft Corporation
P. H. Adams	National Aeronautic Association of the United States of America
R. S. Barnaby	Bureau of Aeronautics, Navy Department
Archibald Black	Garden City, N. Y.
J. F. Hardecker	Naval Aircraft Factory
E. T. Jones	Wright Aeronautical Corporation
Leslie MacDill	Army Air Service
G. J. Mead	Pratt & Whitney Aircraft Co.
Arthur Nutt	Curtiss Aeroplane & Motor Co., Inc.
H. G. Smith	Air Mail Service
R. H. Upson	Aircraft Development Corporation
Edward Wallace	Glenn L. Martin Co.
Marsden Ware	National Advisory Committee for Aeronautics
K. H. White	Chance Vought Aircraft Corporation
T. P. Wright	Curtiss Aeroplane & Motor Co., Inc.

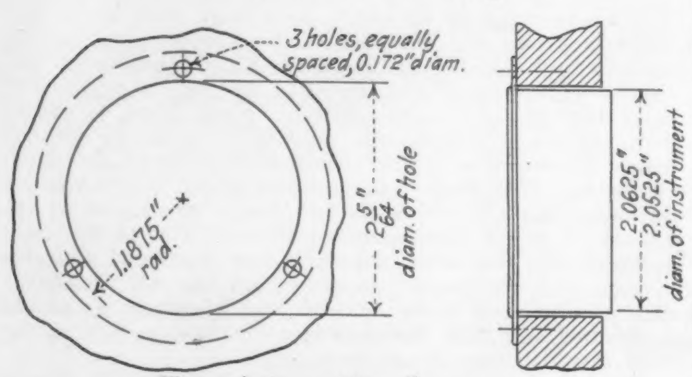
INSTRUMENT MOUNTINGS REVISED

Adoption of 2-In. Size Recommended by Electrical Equipment Division

The Electrical Equipment Division has recommended that the present S.A.E. Standard for Instrument Mountings shown on p. B12 of the S.A.E. HANDBOOK, be extended to include a mounting for the 2-in. diameter clamp-type of instrument on automobile instrument-boards. The report covers the mounting for electrical instruments and for oil, gasoline and pressure and temperature gages, in addition to the present standard flange and screw type of mounting. A survey of present practice indicated that the proposed new mounting will be generally acceptable either where the instrument is mounted directly on the instrument-board or where it is one of a group on a sub-panel. The Division felt that the present standard should be retained as this flange and screw type of instrument is still used in motor-trucks.



Back-of Board or Clamp Mounting



Flanged Screw Mounting

The revised report as recommended by the Electrical Equipment Division is shown in the accompanying illustration.

As a result of discussion at the meeting, work will be continued toward standardizing the size of the terminal studs and the connection on the back of oil, gasoline, pressure and temperature gages, probably incorporating the present S.A.E. Recommended Practice for Pressure-Gage Connections shown on p. C45a of the S.A.E. HANDBOOK.

ELECTRICAL EQUIPMENT DIVISION PERSONNEL

T. L. Lee, *Chairman*	North East Electric Co.
B. M. Leece, *Vice-Chairman*	Leece-Neville Co.
Azel Ames	Kerite Insulated Wire & Cable Co., Inc.
F. W. Andrew	Andalusia, Pa.
W. B. Churcher	White Motor Co.
C. F. Gilchrist	Electric Auto-Lite Co.
W. S. Haggott	Packard Electric Co.
A. D. T. Libby	Representing Automotive Electric Association
W. P. Loudon	Dayton Engineering Laboratories Co.
D. M. Pierson	Dodge Bros.
F. H. Prescott	Remy Electric Co.
B. M. Smarr	General Motors Corporation
T. E. Wagar	Studebaker Corporation of America

DISTRIBUTOR-ROTOR ELECTRODE

New Terms To Be Included in Electrical Equipment Nomenclature

The Society recently received an inquiry as to the proper name for that part of a timer-distributor rotor from which the high-tension spark jumps to the cap segments in the type of timer-distributor rotor that has a small electrode rather than a carbon brush. The term suggested for adoption was Distributor-Rotor Electrode and, as the members of the Division considered this as descriptive as any that could be used, the Division has recommended that it be added to the S.A.E. Nomenclature, immediately following the term Distributor-Rotor Brush on p. K7 of the S.A.E. HANDBOOK.

The Subdivision on Ground-Return Wiring System has as a result of its work proposed that the term Load-Limit Controller be included in the S.A.E. Nomenclature to designate the protective element that is frequently used in the generator circuit to prevent overload on a voltage-regulated shunt-wound generator. The Division also recommends that this term be inserted in the standard nomenclature, immediately following the term Current-Voltage Regulator on p. K9 of the S.A.E. HANDBOOK.

GROUND-RETURN WIRING SYSTEM

Report on Revised Standard Approved for Lighting and Starting Circuits

The Electrical Equipment Division, at its meeting in Detroit on Nov. 9, 1926, reviewed the report of the Subdivision that was prepared as a revision of the present S.A.E. Standard, p. B28 of the S.A.E. HANDBOOK. This work has been in progress for some time and the latest revision is the result of general circulating of the Subdivision's preliminary re-

port. The report as approved by the Electrical Equipment Division follows:

The systems of installation commonly known as two-wire or single-wire, shall be termed respectively insulated return and ground-return systems.

Installations in which the chassis frame is used as part of the return circuit shall be considered as ground-return systems.

Insulated Cable.—All insulated cable shall conform to the S.A.E. Standard. S.A.E. Standard colors shall be used as a means of identification. Copper terminals on other than starting-motor cables shall be clamped to the insulation and soldered to the conductors. S.A.E. Standard terminals or lugs shall be used on starting-motor cables. Conductor cross-sections shall be not less than the equivalent of No. 16. B. & S. gage and the potential loss at normal load shall not exceed 3 per cent.

Conduit.—Insulated cable shall be protected by metal armor, or unpacked metallic or non-metallic conduit except where otherwise protected or not in contact with metal surfaces. Metallic conduit shall be provided with ferrules having rounded edges at open ends and where it enters junction-boxes. Metallic conduit covering insulated cable leading to a connector shall be soldered inside of a sleeve member of the connector. Non-metallic conduit shall be provided with metal clamps at the entrance to junction-boxes or at connectors. Wires unprotected with conduit shall be cleated at intervals not exceeding 28 in. Wires protected with conduit shall be cleated at intervals of not more than 36 in. with metal clips secured by bolts or wood screws. Staples shall not be used. No wire shall be nearer to the exhaust-pipe than 2 in. A minimum clearance of 1 in. shall be maintained between any running line of conduit or wires and the carbureter, gasoline pipe, gasoline tank, or any moving part. All holes in metal members through which cables not in conduit pass, shall have edges rolled or shall be provided with rubber bushings.

Grounding.—Wherever a conductor is connected to ground it shall be installed so as to be accessible for repair. Ground-return connections shall be made to the chassis frame or to a substantial part which is firmly attached to the frame at two or more places. Ground connections shall be made to the frame by terminals soldered to the conductors. The surfaces on which the terminals make contact shall be clean and free from oxide or paint. The terminals shall be fastened to the frame by screws or bolts. The storage-battery shall be grounded on the positive side by a conductor securely bolted to the frame, not to an attached member. The contact surface on frames should be tinned.

Battery Installation.—The battery shall be installed so that an overflow of electrolyte will not cause appreciable leakage of current. The storage-battery compartment, or metal parts near the storage-battery, shall be painted with an acid resisting paint and shall be provided with openings of a size to provide ample ventilation and drainage. At the point where the live line passes through a metal compartment the cable shall be protected against grounding by an acid and waterproof insulating bushing. Where a battery and a gasoline tank are both placed under the driver's seat they shall be partitioned from each other. Each compartment shall be provided with an independent cover, ventilation and drainage.

Overload Protective Devices.—The current to all low-tension circuits, except starting-motor and ignition circuits, shall pass through protective devices connected to the battery-feed side of switches. The circuits shall be arranged so that the opening of a protective device will not extinguish all the lights. The inspection-lamp cord plug-socket shall be connected to an unprotected live circuit. A permanently connected inspection lamp or accessory cord shall be protected independently of the lighting.

Protection Against Accidental Short-Circuits.—Connecting posts on fuse and junction-blocks shall be enclosed as a protection against accidental short-circuits.

WIRING COLOR-CODE RECOMMENDED

Electrical Equipment Division Approves Subdivision Report for Wire Colors

The first tentative report of the Subdivision that was organized to prepare a color code for electrical wiring in motor-vehicles was published on p. 571 of the June, 1926, issue of THE JOURNAL. Several constructive criticisms of the preliminary report were received and the Subdivision redrafted the proposal and again circularized it. Information in the hands of the Electrical Equipment Division at its meeting on Nov. 9, 1926, indicated that, although there are still some criticisms of the proposal, it meets with general approval and in fact forms a basis for wiring systems recently adopted by some of the automobile companies. The Division felt that the report is now in good form and has recommended it for adoption by the Society as S.A.E. Recommended Practice. The report follows:

PASSENGER-CAR WIRING COLOR-CODE FOR USE WHERE CABLE IS BOUGHT IN COILS

RED (Unprotected Live Wires)

Generator to Cut-Out or Regulator
Cut-Out or Regulator to Ammeter
Ammeter to Battery
Ammeter to Overload Breaker or Fuse
Low-Tension or Primary Ignition
All Other Unprotected Live Wires

YELLOW (Protected Live Wires)

Horn Feed Wire
Signal-Lamp Switch Feed Wire
Body-Lighting Switch Feed Wires
Protective Devices to Lighting Switches
All Other Protected Live Wires

PASSENGER-CAR WIRING COLOR-CODE FOR USE WHERE CABLE IS BOUGHT IN FORM OF HARNESS

RED (Unprotected Live Wires)

Generator to Cut-Out or Regulator
Cut-Out or Regulator to Ammeter
Ammeter to Overload Breaker or Fuse
All Other Unprotected Live Wires

RED WITH YELLOW TRACER

Low-Tension or Primary Ignition

RED WITH BLACK TRACER

Ammeter to Battery

YELLOW (Protected Live Wires)

Horn Feed Wire
Signal-Lamp Switch Feed Wire
Body-Lighting Switch Feed Wires
Protective Devices to Lighting Switches
All Other Protected Live Wires

BROWN WITH BLACK TRACER

Lighting Switch to Junction Block (Parking Lamp)
All Ground Connections (except Battery Ground)

BLACK

Lighting Switch to Tail-Lamp

BLACK WITH RED TRACER

Bright Head-Lamps (or Upper Beam)

GREEN

Dim Head-Lamps (or Lower Beam)
Signal-Lamps (Switch to Lamp)

MOTORCOACH AND TRUCK WIRING COLOR-CODE[1]

RED (Unprotected Live Wires)

Generator to Cut-Out or Regulator
Cut-Out or Regulator to Ammeter
Ammeter to Battery
Ammeter to Overload Breaker or Fuse
Low-Tension or Primary Ignition
All Other Unprotected Live Wires

YELLOW (Protected Live Wires)

Horn Feed Wire
Signal-Lamp Switch Feed Wire
Body-Lighting Switch Feed Wires
Protective Devices to Lighting Switches
All Other Protected Live Wires

BROWN WITH BLACK TRACER

Generator Cut-Out or Regulator to Ground
All Ground Connections (except Battery Ground)

BLACK

Bright Head-Lamps (or Upper Beam)
Body-Lamp Feed Wires (Switch to Lamp)

BLACK WITH RED TRACER

Dim Head-Lamps (or Lower Beam)
Tail-Lamp

GREEN

Signal-Lamp (Switch to Lamp)
Signal-Lamp (to Indicator or Pilot)

[1] It is assumed that motorcoach and motor-truck builders will buy cable in bulk.

FLYWHEEL-HOUSING STANDARD CLARIFIED

Wording of Footnotes Changed by Transmission Division To End Confusion

At the meeting of the Transmission Division, which was held in conjunction with the Engine Division at Detroit on Nov. 11, 1926, the notes in the S.A.E. Standard for Flywheels and Flywheel Housings, p. A1 of the S.A.E. HANDBOOK were revised to read "indicated runout" instead of "indicator reading," inasmuch as the present expression has led to confusion in a number of instances. The other minor correction in these notes that was approved was omitting the cipher in the fourth place of decimals so that the notes would be interpreted as indicating measured accuracy within three decimal places rather than four.

ENGINE DIVISION PERSONNEL

The list of the members of the Engine Division is as follows:

A. F. Milbrath, *Chairman*	Wisconsin Motor Mfg. Co.
J. B. Fisher, *Vice-Chairman*	Waukesha Motor Co.
R. J. Broege	Buda Co.
L. F. Burger	International Harvester Co.
E. P. Gundry	Buick Motor Co.
J. D. Harris	McCord Radiator & Mfg. Co.
C. B. Jahnke	Fairbanks, Morse & Co.
L. P. Kalb	Continental Motors Corporation
L. S. Keilholtz	Delco-Light Co.
E. T. Larkin	Sterling Engine Co.
E. S. Marks	H. H. Franklin Mfg. Co.
R. C. McWane	Associated Automobile Engine Rebuilders
Arthur Nutt	Curtiss Aeroplane & Motor Co., Inc.
J. C. A. Straub	International Harvester Co.
L. M. Woolson	Packard Motor Car Co.

NEW ENGINE-TESTING FORMS

Proposed Charts Will Be Applicable to All Internal-Combustion Engine Types

The present S.A.E. Standard for Engine-Testing Forms on p. A51 of the S.A.E. HANDBOOK, which was adopted by the Society in 1917 and revised in 1924, is used very extensively although the complete forms are suitable for only a limited range of engines. The Engine Division has undertaken a complete review of the forms and a rearrangement and extension of them, particularly the curve sheet, so that they can be used more readily for all common types and sizes of internal-combustion engine. A Subdivision was appointed consisting of L. P. Kalb of the Continental Motors Corporation, chairman; L. S. Keilholtz, of the Delco Light Co.; E. T. Larkin, of the Sterling Engine Co.; L. K. Marshall, of the General Motors Corporation Research Laboratories; A. W. Pope, Jr., of the Waukesha Motor Co.; and Prof. O. W. Sjogren, of the University of Nebraska. This Subdivision made a careful study of the present forms and proposed a number of changes in and extensions of them. The report of the Subdivision was carefully reviewed by the Engine Division at its meeting in Detroit on Nov. 11, and approved with the recommendation that the following revisions be made in the present S.A.E. Standard.

RULES AND DIRECTIONS—SHEET A

In the fourth paragraph following the heading General Rules and Directions, the word "torque" in next to the last line is changed to "horsepower."

Under General Rules and Directions, the following paragraph is to be added immediately following the paragraphs under Indicated Horsepower.

> *Correction Factors.*—All results shall be corrected to a standard barometric pressure of 29.92 in. of mercury and a standard temperature of 520 deg. fahr. absolute, which is the same as 60 deg. fahr. These corrections are to be made by using the correction formula.

The form of the correction formula printed at the end of Sheet A is to be clarified by extending the radical sign over the entire term T_o/T_s and changing the wording in the key to P_s to read

> Standard barometric pressure of 29.92 in. of mercury.

The key to T_s is changed to read

> Standard absolute temperature of 520 deg. fahr.

The correction formula and key are to be transposed to follow the new paragraph on Correction Factors.

In the first paragraph following the subheading Specification Sheet, the words "or kerosene" are omitted at the end of the fifth line and the speeds "100 to 120 r.p.m." in the last line of the paragraph are changed to read "speed of maximum torque."

SPECIFICATION SHEET—B

No changes are proposed in this sheet.

LOG SHEET—C

This sheet is to be turned on the page so that it will read from top to bottom uniformly with the other three sheets. Provision for recording the room temperature and barometric reading has been made in the body of the sheet and these items should therefore be omitted from the upper right-hand corner of the sheet. The following changes and additions have been made to items under Brake Horsepower and Fuel Consumption.

> Add the word "Observed" to "Torque in Lb-Ft."
> Add the term "Torque Corrected" immediately following the above.

Add "Observed" to the item "Brake Horsepower."
Add "Brake Horsepower Corrected" immediately following the above.
Add "Observed" to the term "Indicated Horsepower."
Add "Indicated Horsepower Corrected" immediately following the above.
Immediately following "Temp. Jacket Water—Out" add the three items in the order given.

(1) Temperature of oil in, deg. fahr.
(2) Temperature of oil out, deg. fahr.
(3) Oil pressure, lb.

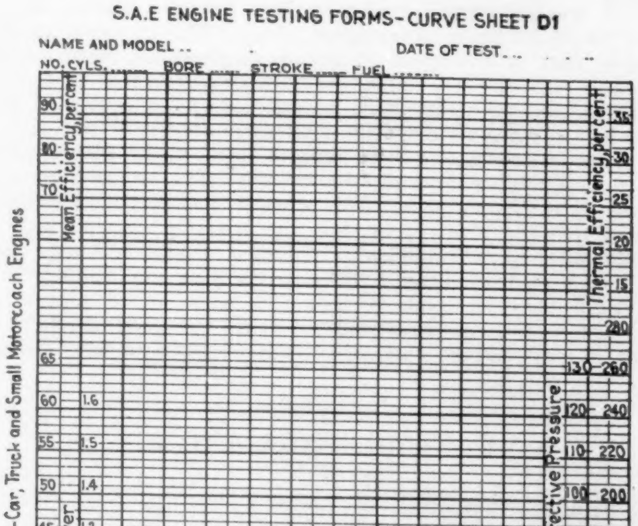
Immediately following "Temp. Air to Carb." include the item "Room Temp. deg. fahr."
Immediately following the last item "Thermal Eff. re B. Hp. add the term "Barometer, In. Hg."
Leave two blank spaces following this group of items and two or more blank spaces following the group of items under "Friction HP."
Omit from the bottom of the sheet the sentence beginning "If readings are corrected to 60 deg. fahr."

Curve Sheets—D1 to D6

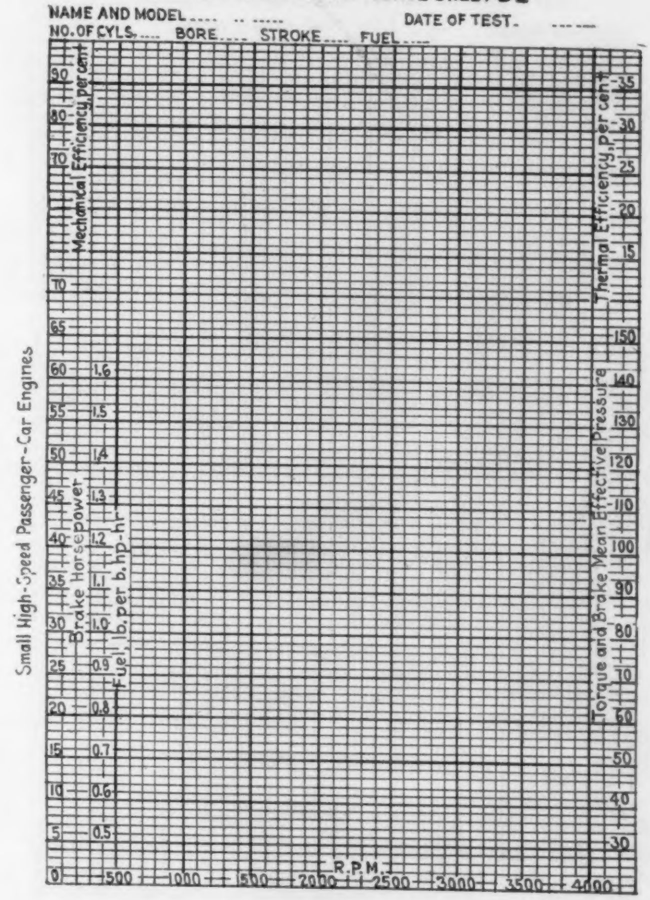
Experience in plotting engine curves on the present Standard curve-sheet has shown that the sheet is inadequate and poorly arranged with regard to the ordinate scales and the Subdivision therefore arranged five general classifications of engine and a new curve-sheet for each. When the subject was discussed by the Engine Division, it was felt that a still larger type of engine than provided for by the Subdivision should be included in the classification for high-horsepower airplane engines and the heavier-duty type of engine for industrial and motive-power applications. It was therefore decided to add still another curve-sheet, making six sheets in all, classified as follows:

D1—Low-speed passenger-car, truck and small motorcoach engines
D2—Small high-speed passenger-car engines

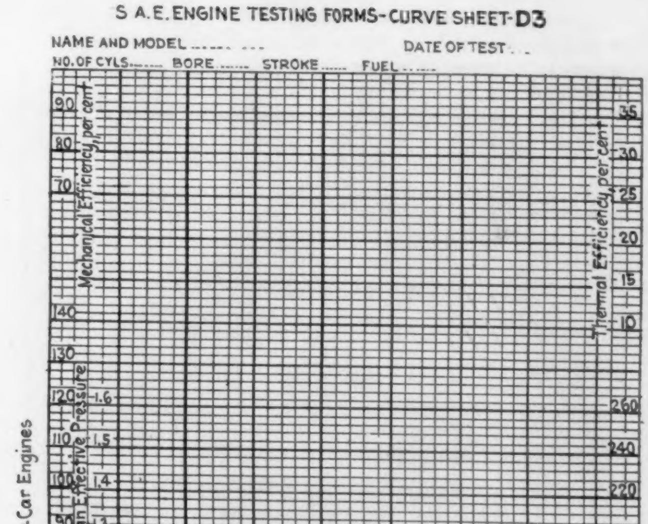
S.A.E ENGINE TESTING FORMS-CURVE SHEET D1



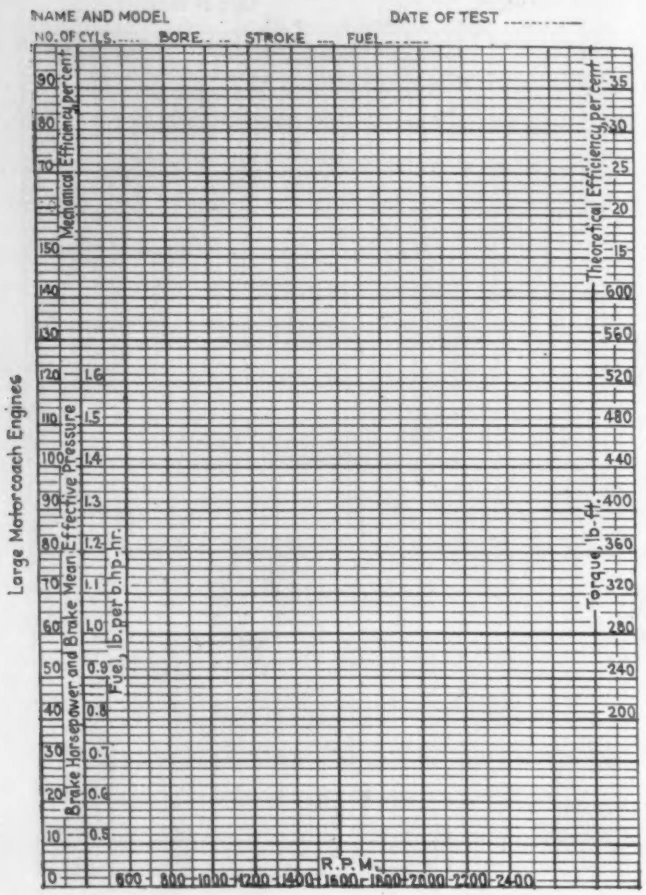
S.A.E. ENGINE TESTING FORMS- CURVE SHEET-D2



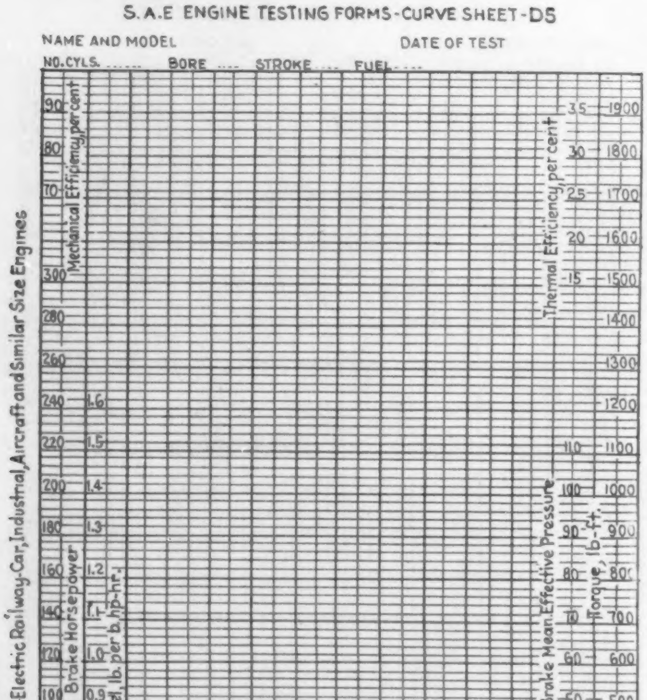
S A.E. ENGINE TESTING FORMS-CURVE SHEET-D3



S. A. E. ENGINE TESTING FORMS - CURVE SHEET-D4



S. A. E. ENGINE TESTING FORMS - CURVE SHEET - D5



D3—Large passenger-car engines
D4—Large motorcoach engines
D5—Gasoline-electric rail-car, industrial, aircraft, and similar size engines.

D6—The Subdivision felt that further data on this class of engine should be had before preparing this sheet and these will be secured as soon as possible. The form of the sheet will follow that of the others and it is hoped it can be ready at the time of the Standards Committee Meeting

The Subdivision in reporting to the Division felt that the temperature and barometric correction-formula should be applied to the indicated horsepower rather than the brake horsepower but that the corrections for temperature and barometer in this connection should be studied further before a change shall be definitely recommended. It was also felt that a chart, suggested forms of which have already been submitted to the Subdivision, which will facilitate finding corrected brake-horsepower from observed brake-horsepower knowing the barometer, temperature and mechanical efficiency, would be valuable to include in the forms, but this matter also was left by the Division for further study. The Division expects to take these points up with the engine and car builders, as well as with the Bureau of Standards which has done considerable work in this conneciton.

CRANKCASE DRAIN-PLUGS REVISED

Engine Division Adds Two Sizes to Present Recommended Practice and Drops One

Following a survey of practice with regard to the dimensions of shanks on crankcase drain-plugs, the Engine Division has recommended that the present S.A.E. Recommended Practice, p. A3b of the S.A.E. HANDBOOK, be completed by including a 1⅛-in. hexagon shank for the ⅞-in.-18 straight-threaded plugs and a 7/16-in. square shank for the ½-in. taper pipe-thread plugs. The Division appreciated that in a number of instances plugs are used which have a wrench-hole instead of a shank, but felt that this type of plug is undesirable because of the difficulty of removing it without a special wrench and because the usual practice of mechanics is to remove these plugs with a pair of pliers, thus destroying the thread. The suggestion was made that the recommended practice included a standard connection for drain-cocks, but the Division decided not to make such a recommendation because drain-cocks are too liable to be broken off by obstructions on the road. The Division also recommends that the ⅜-in. taper pipe-plug be omitted as it is considered too small for good practice in that it restricts the rush of oil out of the crankcase when draining it.

S.A.E. STEELS 2015 AND 2115

Division Confirmation of Letter-Ballot on Nickel-Steel Compositions

The Iron and Steel Division at a meeting held in the offices of the Society on Nov. 23, 1926, confirmed the letter-ballot approving the proposed compositions of S.A.E. Steels 2015 and 2115, details of which were printed in the table of proposed S.A.E. Standard iron and steel chemical compositions on p. 451 of the November issue of THE JOURNAL. These steels contain 0.50 and 1.50 per cent of nickel respectively and have been approved with the understanding that they will be subject to consideration of the phosphorus and sulphur content which will be investigated by a subdivision to be appointed by Chairman Watson for the purpose of preparing an exhaustive report on phosphorous and sulphur content in S.A.E. Steels.

The Iron and Steel Division, the personnel of which is printed below, therefore recommends that the following

nickel-steel compositions be included with those printed on p. D5 of the S.A.E. HANDBOOK as S.A.E. Standard.

S.A.E. Steel	2015	2115
Carbon Range, per cent	0.10–0.20	0.10–0.20
Manganese Range, per cent	0.30–0.60	0.30–0.60
Phosphorus Maximum, per cent	0.040	0.040
Sulphur Maximum, per cent	0.045	0.045
Silicon Range, per cent	0.15–0.30	0.15–0.30
Nickel Range, per cent	0.40–0.60	1.25–1.75

IRON AND STEEL DIVISION PERSONNEL

J. M. Watson, *Chairman*	Hupp Motor Car Corporation
J. H. Nelson, *Vice-Chairman*	Wyman-Gordon Co.
J. R. Adams	Midvale Co.
A. L. Boegehold	General Motors Corporation Research Laboratories
H. T. Chandler	Vanadium Corporation of America
J. D. Cutter	Climax Molybdenum Co.
A. H. d'Arcambal	Pratt & Whitney Co.
C. N. Dawe	Vanadium Corporation of America
B. H. DeLong	Carpenter Steel Co.
F. P. Gilligan	Henry Souther Engineering Corporation
H. L. Greene	Willys-Overland Co.
E. J. Janitzky	Illinois Steel Co.
J. B. Johnson	Air Corps
J. A. Matthews	Crucible Steel Co. of America
G. L. Norris	Vanadium Corporation of America
W. C. Peterson	Donner Steel Co.
Representative	Bureau of Engineering, Navy Department
S. P. Rockwell	Stanley P. Rockwell Co.
C. F. W. Rys	Carnegie Steel Co.
R. B. Schenck	Buick Motor Co.
M. H. Schmid	United Alloy Steel Corporation
R. H. Sherry	Evanston, Ill.
W. R. Shimer	Bethlehem Steel Co.
T. H. Wickenden	International Nickel Co.

CHANGE IN MANGANESE LIMITS

Iron and Steel Division Revises Specification for S. A. E. Steel 1046

S. A. E. Steel 1046, which was added to the Society's steel specifications at the request of the American Gear Manufacturers Association and is used solely by gear manufacturers, was the subject of discussion at the last meeting of the Iron and Steel Division held in the offices of the Society on Nov. 23, 1926.

The use of this steel for street-car gears has developed the need for revised limits for the manganese content and the combined carbon and manganese-content.

S. P. Rockwell, as representative of the American Gear Manufacturers Association, after reading a summary of the opinions of gear users favoring the change, requested that the Division consider recommending manganese-content limits of 0.40 to 0.60 per cent and the addition of a specification calling for a combined carbon and manganese-content limit of 0.85 to 1.05 per cent. This brought up a question of policy regarding the admission to S. A. E. Steel Standards of specifications for a steel that is not primarily intended for automotive purposes. In this instance the steel under consideration is not an automotive steel, and the Division felt that it was inadvisable, from a precedent standpoint, to include it in the S.A.E. Standard. To change the specification as requested would introduce a limit on the combined content of carbon and manganese, which is a factor not appearing elsewhere in any of the S. A. E. Steel Specifications. It was, therefore, voted to recommend a change in the manganese range only, from 0.30 to 0.50 per cent to 0.40 to 0.60. The proposed standard follows:

Carbon Range, per cent	0.40–0.50
Manganese Range, per cent	0.40–0.60
Phosphorous Maximum, per cent	0.045
Sulphur Maximum, per cent	0.050

The phosphorus and the sulphur content are subject to consideration by a subdivision to be appointed for the investigation of phosphorus and sulphur limits in S. A. E. Steels.

ROCKWELL HARDNESS-TEST APPROVED

Present S.A.E. Specifications Cover Only Brinell and Shore Hardness-Tests

At the May, 1926, meeting of the Iron and Steel Division the desirability of including the Rockwell Hardness-Test in the present S.A.E. Recommended Practice for Hardness Tests, which includes the Brinell and Shore methods of testing, was recognized by the approval of a report submitted by the Subdivision on Hardness Tests, consisting of J. H. Nelson, of the Wyman-Gordon Co., chairman; L. A. Danse, of the Cadillac Motor Car Co.; H. L. Greene, of the Willys-Overland Co., and S. P. Rockwell, of the Stanley P. Rockwell Co.

The Division at its meeting in New York City on Nov. 23, 1926, confirmed the letter-ballot approving the recommendation with a few minor changes in wording made by Mr. Rockwell since then, and now submits the report for approval as S.A.E. Recommended Practice under Part VIII—Hardness Tests—Iron and Steel Specifications, p. D15 of the S.A.E. HANDBOOK.

ROCKWELL HARDNESS-TEST

Principle of Test.—The Rockwell tester measures hardness by determining the depth of penetration of a steel ball or diamond cone (Brale) in the material being tested under conditions of load application. In elevating the work to testing position, the machine first applies a preliminary, or minor, load of 10 kg. (22.046 lb.), which clamps and seats the piece being tested and allows the penetrator to break through the light scale and come in contact with the true material beneath. The direct-reading dial is set to zero penetration, directly after due application of the minor load. With the minor load applied and a zero penetration reading known, a final or major load is applied. This forces the penetrator into the metal being tested. The indication at this time on the direct-reading dial comprises not alone the depth of penetration but also the deformation of the testing-machine frame and components, and the work undergoing test. The true Rockwell hardness is read directly from the dial, after removing the major load. This automatically reinstates the 10-kg. (22.046-lb.) minor load and removes the deformation of the machine and part under test. The result is the permanent deformation of the tested metal due to the penetrator and its major load. The Rockwell hardness-readings are based on the depth to which the major load forces the penetrator below the point it was forced previously by the minor load.

Preparation of Surfaces.—Concordant results are dependent on surface roughness being much less than the size of the impression. Surfaces that are ridged perceptibly to the eye by rough grinding or machining offer unequal support to the penetrator. The degree of surface preparation then depends to some extent on the requirements of testing, whether they be production or research.

Thickness of Specimens.—Practically any thickness in excess of 0.020 in. can be tested for comparative hardness. For true hardness, the piece must be of such thickness that the under surface of the specimen, after testing, does not show a point of compression. The minimum possible thickness of any specimen varies according to the hardness, the load applied and the kind of test-point or penetrator used. The hardest steels

give true hardness-readings if over 0.027 in. in thickness.

Curved Surfaces.—The true hardness of parts having curved surfaces can be found if the radius of curvature of the surface at the point tested is 3/16 in. or more. If the radius is less than 3/16 in., only comparative results can be had. Data for hardness-tests on a highly curved surface should be accompanied by a statement of the radius of curvature. In testing small rounds, the effect of curvature can be eliminated by making a small flat-spot on the specimen.

Rockwell Scales.—Since two loads and two penetrators are used, two scales of hardness are placed on the dial of the machine. The black divisions and the letter C apply when the 120-deg. diamond-cone penetrator or Brale and the 150-kg. (330.693-lb.) load are employed. The red divisions and the letter B apply when the 100-kg. (220.462-lb.) load and the 1/16-in. steel-ball penetrator are employed. All data should be prefixed by a letter showing whether the values are on the C or the B scale. Other loads and penetrators can be used, but only comparative results can be obtained. In recording such results all factors relating to them should be clearly stated: the load; the penetrator; the scale used; the thickness of the piece, if very thin; and the curvature of the surface.

Penetrators and Loads.—The Rockwell scale being based on depth measurements, the Brale penetrator gives the more accurate readings on hard metals and the steel-ball on soft metals. Hard metals require the 150-kg. (330.693-lb.) load; the diamond-cone penetrator of 120-deg. angle with the point ground spherically to a definite radius is the standard Rockwell diamond-cone or Brale.

If the readings are below C-20, it is preferable to test the material with the 100-kg. (220.462-lb.) load and the 1/16-in. steel-ball (replaceable) and use the B scale. Readings on the B scale are not generally taken higher than B-100, when it is advisable to change to the C scale with the 150-kg. (330.693-lb.) load and the Brale penetrator.

Measuring hardness greater than B-100 is apt to deform the 1/16-in. steel-ball, thus destroying its usefulness for testing softer materials.

Each tester is provided with means for regulating the rate of application of the load, for which the standard speed is 5 sec. for production-testing on steel. A speed of 3 sec. can be used without appreciably affecting the readings. Metals that flow readily under pressure, like zinc, should be tested under a specified rate and the duration of the application of the load should also be specified.

Placing machines where extreme vibrations are in effect should be avoided as this causes fluctuations of the major load during test and correspondingly low readings.

Chucking the Work.—Parts with much overhang should be suitably supported so that, when the minor load is applied, the work will be held rigidly. The piece being tested, while under the pressure of only the minor load, must be secure so that it will not tend to tip at the edge of the anvil and move laterally under the major load.

The surface tested should not vary from the horizontal with reference to the vertical axis of the penetrator more than 7½ deg. In testing cylindrical pieces, such as wire, an anvil provided with a V-notch should be used and the point of penetration applied vertically over the axis of the piece. The under-surface of the piece, where it rests on the anvil, must be free from scale or burrs that might collapse or flatten during the application of the major load.

On truly homogeneous material sub-low readings are due to burrs on the under-surface of the material tested, causing a movement of the material on its chuck during application of the major load. Assuming that the machine checks correctly on its test blocks, too high readings can be secured only by testing on a spot previously tested, by testing too thin sections or by the penetrator being supported by two ridges caused by poor surface preparation, the last-named being a minor consideration in production testing.

In testing tubing or any shape that will deform under test, a blunt nose should be used in place of the penetrator, and the test conducted in the regular way to determine if the specimen takes a permanent set, that would invalidate the hardness-test and necessitate consideration of another manner of chucking.

Homogeneity of Metals.—The Rockwell tester measures the hardness of the specimen at the point of penetration, but the reading is also influenced by the hardness of the material under the impression. For example, in testing the hardness of carbon steels, when using the Brale penetrator and the 150-kg. (330.693-lb.) load, the actual depth of penetration is about 0.0027 in. while the effects of penetration extend to about 0.027 in., and if any softer layer is located within 0.027 in. of the impression, the impression will be deeper and the apparent hardness less. Therefore, due regard for this condition must be given when testing material with a superficial hardness, such as cyanided or nitrogenized surfaces.

When testing bar stock and forgings, it will be found advisable to remove enough of the surface metal by machining or grinding so that the penetrator will test the true metal underneath, and the hardness reading will not be affected by the heavy scale and decarburization usually present.

Materials, such as cast iron with graphite particles and some non-ferrous materials whose crystalline aggregates are greater than the area of the penetrator, must be tested, if possible, with a penetrator of sufficient size to overcome local or grain hardness in order to secure mass hardness.

Conversion to Other Methods of Hardness-Testing.—Care should be used in converting readings from one method of hardness-testing to another. The factors that cause erroneous conversions are:

Lack of true homogeneity in metals.

Different depths from the surface at which various hardness-tests operate.

Different physical characteristics of various testing devices and the properties measured.

Different properties that are assumed by different metals under cold-working.

No one conversion-table is practical for close conversion on all metals. Tables and formulas have been compiled for some materials that can be used as guides. When conversion is necessary, it is advisable to make such conversion for each particular type of metal and each condition of heat-treatment or mechanical work.

MOLYBDENUM STEELS HEAT-TREATMENT

S.A.E. Steels 4130 and 4140 Covered by Iron and Steel Division Report

At the May, 1926, meeting of the Iron and Steel Division members in Bethlehem, the accompanying report of the Subdivision on Physical-Property Charts covering notes and instructions for S. A. E. Molybdenum Steels 4130 and 4140 was approved.

NOTES AND INSTRUCTIONS FOR MOLYBDENUM STEELS
S.A.E. STEEL 4130

These notes are not to be considered in any way a part of the standard specifications for S.A.E. Steels. They are added solely for the information of users of the steels and the guidance of purchasers in the selection of proper materials for different purposes. They

should not be incorporated in the customer's specifications when ordering steel.

CHEMICAL COMPOSITION IN PERCENTAGE

Carbon	0.25 - 0.35
Manganese	0.40 - 0.70
Phosphorus	0.040 max.
Sulphur	0.045 max.
Chromium	0.50 - 0.80
Molybdenum	0.15 - 0.25

This steel may be used interchangeably with Steels 2330, 3130, 3135, and 6130 for heat-treated automotive forgings requiring greater strength and toughness than are obtainable with plain carbon-steels.

As the structure of forgings is less uniform than that of bar stock in most cases, either in the individual piece or between different pieces, normalizing as in Heat-Treatment VII is recommended as the preliminary treatment for all forgings, but the desired physical properties of bar stock can be obtained generally without normalizing, as in Heat-Treatment VI.

Heat-Treatment 4130—*VI*

(1) Heat to 1550 to 1650 deg. fahr.
(2) Quench
(3) Draw to required hardness

Heat-Treatment 4130—*VII*

(1) Normalize at 1650 to 1750 deg. fahr.
(2) Reheat to 1550 to 1650 deg. fahr.
(3) Quench
(4) Draw to required hardness

For forgings that are to be machined after heat-treatment, Heat-Treatment VII is recommended.

S.A.E. STEEL 4140

These notes are not to be considered in any way part of the standard specifications for S.A.E. Steels. They are added solely for the information of users of the steels and the guidance of purchasers in the selection of proper materials for different purposes. They should not be incorporated in the customer's specifications when ordering steel.

CHEMICAL COMPOSITION IN PERCENTAGE

Carbon	0.35 - 0.45
Manganese	0.40 - 0.70
Phosphorus	0.04 max.
Sulphur	0.045 max.
Chromium	0.80 - 1.10
Molybdenum	0.15 - 0.25

This steel may be used interchangeably with Steels 2340, 3140 and 6140 for heat-treated automotive forgings requiring greater strength and toughness than are obtainable with plain carbon-steels.

As the structure of forgings is less uniform than that of bar stock in most cases, either in the individual piece or between different pieces, normalizing as in Heat-Treatment VII is recommended as the preliminary treatment for all forgings, but the desired physical properties of bar stock can be obtained generally without normalizing, as in Heat-Treatment VI.

Heat-Treatment 4140—*VI*

(1) Heat to 1525 to 1625 deg. fahr.
(2) Quench in oil
(3) Draw to required hardness.

For all general requirements Heat-Treatment VII is recommended.

Heat-Treatment 4140—*VII*

(1) Normalize at 1650 to 1750 deg. fahr.
(2) Reheat to 1525 to 1625 deg. fahr.
(3) Quench in oil
(4) Draw to required hardness

Parts that are to be machined after forging and before heat-treatment, Heat-Treatment VIII is recommended.

Heat-Treatment 4140—*VIII*

(1) Normalize at 1650 to 1750 deg. fahr.
(2) Reheat to 1250 to 1350 deg. fahr.
(3) Cool slowly
(4) Machine
(5) Reheat to 1525 to 1625 deg. fahr.
(6) Quench in oil
(7) Draw to required hardness.

At a meeting of the Iron and Steel Division held in the offices of the Society on Nov. 23, 1926, the Subdivision on Physical-Property Charts recommended, for approval as general information, notes and heat-treatments for S.A.E. Steel 4615. The use of molybdenum steel 4150 has been insufficient to permit the recommendation of similar data for it and it was, accordingly, decided to recommend heat-treatments for S.A.E. Steel 4615 only at this time.

The report of the Subdivision as approved by the Division follows:

S.A.E. STEEL 4615

These notes are not to be considered in any way a part of the standard specifications for S.A.E. Steels. They are added solely for the information of users of the steels and the guidance of purchasers in the selection of proper materials for different purposes. They should not be incorporated in the customer's specifications when ordering steel.

CHEMICAL COMPOSITION IN PERCENTAGE

Carbon	0.10 - 0.20
Manganese	0.30 - 0.50
Phosphorus	0.04 max.
Sulphur	0.045 max.
Nickel	1.25 - 1.75
Molybdenum	0.20 - 0.30

This steel is intended primarily for case-hardening. When maximum hardness of the case and maximum refinement of both the case and the core are desired, distortion being unimportant, Heat-Treatment V should be used. When hardness, refinement of the case and the least possible distortion are required, Heat-Treatment IV should be used.

When this steel is used for gears for which a high degree of accuracy and considerable strength are required, it is recommended that the carburizing operation be preceded by normalizing at from 1650 to 1750 deg. fahr., which will improve the structure and tend to reduce the distortion caused by subsequent treatments.

Heat-Treatment 4615—*IV*

(1) Carburize at 1600 to 1650 deg. fahr.
(2) Cool in box
(3) Reheat to 1475 to 1525 deg. fahr.
(4) Quench
(5) Draw at 250 to 500 deg. fahr.

Heat-Treatment 4615—*V*

(1) Carburize at 1600 to 1650 deg. fahr.
(2) Cool in box
(3) Reheat to 1525 to 1575 deg. fahr.
(4) Quench in oil
(5) Reheat to 1375 to 1425 deg. fahr.
(6) Quench
(7) Draw at 250 to 500 deg. fahr.

HEAD-LAMP STRENGTH AND RIGIDITY

Tests Proposed for Adoption as Recommended Practice by Lighting Division

At its Semi-Annual Meeting last June the Society adopted specifications governing the construction of head-lamps, as given in the S.A.E. Recommended Practice on p. B9 of the S.A.E. HANDBOOK. At that time the report included a test for the rigidity of the mounting of head-lamps (Item 10) and a reflection factor (Item 13), but these were not included in the report as adopted as it was felt they should be given further study. The Subdivision subsequently made a further canvass of the industry relative to these items and submitted its report at the meeting of the Lighting Division held in Detroit on Nov. 8, 1926.

With regard to the specification for the rigidity of head-lamp mounting, it was felt that two tests should be considered, the first bearing on the rigidity of the head-lamp shell with regard to its mounting device when firmly held, and the second with regard to the head-lamp when mounted on a motor-vehicle in accordance with standard practice.

At the meeting the statement was made that actual tests had been made along the lines proposed by the Subdivision, and that these were considered adequate. The Lighting Division therefore recommends that the following paragraphs be adopted and included in the present S.A.E. Recommended Practice.

> With the head-lamp mounting firmly attached to a fixed support, there shall be no permanent distortion of the head-lamp as measured by the deflection of the beam, after a steady pressure of 75 lb. is applied for 1 min. to the upper edge of the door in a direction parallel with the head-lamp axis.
>
> With the head-lamp mounted on the car in accordance with standard practice, there shall be no permanent deflection of the head-lamp beam due to distortion of the head-lamp mounting after applying a steady pressure of 75 lb. for 1 min. at the upper edge of the door in a direction parallel with the head-lamp axis.

With regard to writing a definite specification for the reflection factor of head-lamp reflectors, the Subdivision reported that considerable work had been done toward working out some satisfactory means of specifying what this factor should be but that, due to the many variable factors involved, the various kinds of reflecting surface that are available and the many operating conditions that would have to be considered, to write a definite specification without having much more information and more unanimity of opinion bearing on this subject hardly seems possible. No definite action was therefore taken on it by the Division for the time being.

LIGHTING DIVISION PERSONNEL

The list of the members of the Lighting Division is as follows:

C. A. Michel, *Chairman*	Guide Motor Lamp Mfg. Co.
J. H. Hunt, *Vice-Chairman*	General Motors Corporation Research Laboratories
R. E. Carlson	Edison Lamp Works of the General Electric Co.
A. W. Devine	Department of Public Works, Commonwealth of Massachusetts
G. P. Doll	Thomas J. Corcoran Lamp Co.
R. N. Falge	National Lamp Works
C. E. Godley	Edmunds & Jones Corporation
D. A. Harper	Tung-Sol Lamp Works
P. J. Kent	Chrysler Corporation
A. R. Lewellen	Chevrolet Motor Co.
D. M. Pierson	Dodge Bros.
L. C. Porter	Edison Lamp Works of the General Electric Co.
E. S. Preston	Chicago Electric Mfg. Co.
C. D. Ryder	Cincinnati Victor Co.
A. J. Scaife	White Motor Co.
C. L. Sheppy	Pierce-Arrow Motor Car Co.
B. M. Smarr	General Motors Corporation
T. E. Wagar	Studebaker Corporation of America

BASES, SOCKETS AND PLUGS

Lighting Division Recommends Revision of Tolerances in Present Standard

In discussing electric incandescent lamps at the meeting of the Lighting Division in Detroit on Nov. 8, 1926, it was pointed out that the tolerances for the distance from the pin in the base to the end of the soldered tip is specified to limits of from 0.240 to 0.299 in. and that, inasmuch as no manufacturers attempt to gage this dimension so accurately as indicated by the upper limit, this dimension might better be specified in the even decimal 0.300 in. This change will also conform to the corresponding dimension on electric incandescent lamps as reported in this issue of THE JOURNAL. The Division therefore recommends that this change be made in the S.A.E. Standard for Bases, Sockets and Plugs on pages B5, B5a and B5b of the S.A.E. HANDBOOK, together with changing the maximum limit on p. B5 from the center of the pin to the soldered tip from 0.219 to 0.220 in.

NEW STANDARD FOR SIGNAL-LAMPS

Subdivision Report on Recommended Practice Approved by Lighting Division

At the meeting of the Lighting Division in May, 1925, the Division was requested to consider standardization of signal-lamps inasmuch as they had come into general use in great variety. The feeling then was that if a standard could be formulated it would clarify the situation that was developing in connection with the State regulation of motor-vehicles. A Subdivision was appointed, consisting of C. D. Ryder, of the Cincinnati-Victor Co., chairman; C. E. Godley, of the Edmunds & Jones Corporation; and H. H. Magdsick, of the National Lamp Works of the General Electric Co. It investigated the subject and has reported to the Division. The Automobile Lighting Association, which was recently organized and is particularly interested in lamp standardization, has cooperated with the Subdivision in preparing its report. The Lighting Division at its meeting in Detroit on Nov. 8, 1926, reviewed the report and approved it with a few minor modfications in wording to make it conform with other lamp standards of the Society. The Division recommends that the report be adopted as S.A.E. Recommended Practice.

SIGNAL-LAMPS

Signal-lamps, other than those embodying mechanical motion, shall be tested singly and shall meet the following requirements as to light intensity and distribution:

Visibility.—Signal-lamps shall indicate the driver's intention to diminish the speed of, or to stop or to change the direction of a motor-vehicle, by displaying a red light sufficiently bright to attract attention in normal sunlight at a distance of 100 ft. to the rear and 45 deg. to the right and to the left of the vehicle. But signal-lamps shall not project a dangerously glaring or dazzling light.

Light Intensity.—On a line perpendicular to the center of the lamp-face the minimum average brightness shall be 2 cp. per sq. in. over the minimum illuminated area of 3½ sq. in.

At all points at an angle of 30 deg. to the perpendicular to the center of the lamp-face, the minimum average brightness shall be 0.15 cp. per sq. in. over the minimum illuminated area of 3½ sq. in.

In no direction shall the intensity be more than 25 apparent cp.

The Division also recommends that the signal-lamp mounting be the same as the S.A.E. Recommended Practice for Tail-Lamp Mounting, p. B2 of the S.A.E. HANDBOOK, which is by two ¼ in.-20 bolts 2 in. apart extending ⅝ in. from the back of the lamp and on a line horizontal with the plug-socket.

During the discussion at the Division meeting some question arose with regard to whether only a red light should be required or a yellow or amber light should be included in the specification. Inasmuch as most of the signal-lamps in use have the red light and as having practice as nearly uniform as possible is very desirable, the Division felt that the standard as adopted by the Society should indicate the use of but one color.

Controversy has arisen regarding the wording on signal-lamp glasses. The Subdivision is not entirely in agreement on this subject and recommends that the subject not be mentioned in the proposed S.A.E. Recommended Practice. Neither does the Subdivision believe that standardization should be attempted with respect to the position of the signal-lamp relative to the tail-lamp.

REVISED TAIL-LAMP SPECIFICATIONS

Lighting Division Recommendations on Construction and Illumination

When the Lighting Division discussed the subject of signal-lamps as reported elsewhere in this issue of THE JOURNAL, the subject of tail-lamps was also considered for the same general reasons and also in amplification of the present S.A.E. Recommended Practice for Tail-Lamp Illumination as printed on p. B8e of the S.A.E. HANDBOOK. The report of the Subdivision that was considered by the Lighting Division at its meeting in Detroit on Nov. 8, 1926, embodies the requirements of the present S.A.E. Recommended Practice and amplifies it with the exception of those parts of the present recommended practice that do not bear on the technical specification. At the meeting of the Division the question of specifying that the tail-lamp give an amber light was also discussed at considerable length in view of the recent agitation to change current practice. It was felt that the ruby lights should be retained, and the Lighting Division therefore recommends the following for adoption in revision of the present S.A.E. Recommended Practice for Tail-Lamp Illumination:

TAIL-LAMPS

Tail-lamps shall be tested singly and shall meet the following requirements as to construction, light intensity and distribution:

Visibility.—Tail-lamps shall display a ruby light at night, plainly visible under normal atmospheric conditions at a distance of 500 ft. from the rear of the vehicle and shall so illuminate the number-plate carried at the rear of such vehicle as to make it legible at a distance of 50 ft. from the rear of the vehicle.

Construction.—The lamps shall be weather and dustproof and constructed so as to withstand the shock and vibration to which they are ordinarily subjected in use.

Light Intensity.—Tail-lamps shall emit a ruby light which on a line perpendicular to the center of the lamp-face shall be not less than 0.10 apparent cp. and which in all directions at 30 deg. to the perpendicular to the center of the lamp-face shall be not less than 0.05 apparent cp. In no direction shall there be more than 5 apparent cp.

Tail-lamps shall have an opening covered with colorless glass sufficiently large to permit light to cover the entire surface of the registration number-plate, which for the purpose of the test shall be represented by a plane surface 16 in. long by 6½ in. wide in the case of a device for motor-vehicles and 10 in. long by 5 in. wide in the case of a device for motorcycles.

The registration plate holder shall be constructed in such a manner that the major portion of the light incident at any point on the registration plate shall make an angle of not less than 8 deg. with the plane of the plate.

When tested with an incandescent lamp of 2 spherical cp., the illumination as measured on white blotting paper shall be not less than 0.5 foot-candles at any point and the ratio of maximum to minimum intensity shall not exceed 30.

Cut-off of illumination shall not be less than 1½ in. from the plate measured perpendicularly to the plane of the plate at the edge farthest from the lamp.

There must be no unduly bright areas or excessive contrast in the illumination on the registration number-plate.

The Subdivision does not recommend that the American Railway Association's specifications as to color and light transmission be adopted, because of the prohibitive cost of glass so made. The suggestion was made that the Association's specifications might be adopted as to color only, but the Subdivision stated that it does not know of any simple test for determining color.

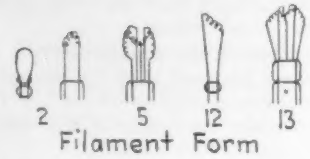
FOCUSING TYPE INCANDESCENT LAMPS

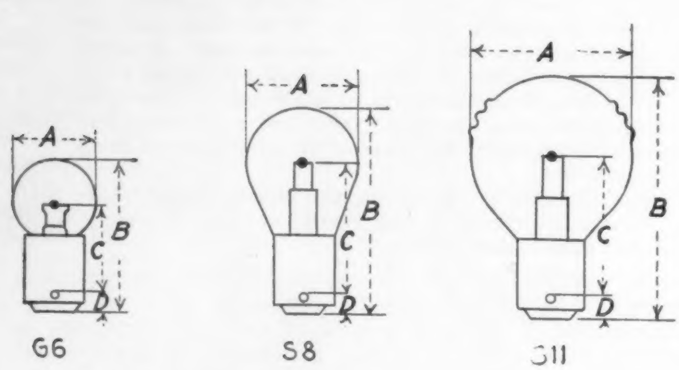
Revision of Present S.A.E. Standard Recommended by Lighting Division

Last spring a subdivision of the Lighting Division was appointed to review the S.A.E. Standard on Electric Incandescent Lamps of the focusing type, p. B4 of the S.A.E. HANDBOOK, in view of recent developments in lamps and the introduction of the depressed-beam or two-filament type of head-lamp. A thorough study of the subject has been made by the Subdivision which reported its recommendations at the meeting of the Lighting Division, held in Detroit on Nov. 8, 1926. It was also reported that the members of the Division who had previously cast a letter-ballot on the report had approved it. Several suggestions that had been submitted by other members were discussed and included in the recommendations. One of the principal points in connection with the report that was considered related to converting the tolerance of the light-center length and axial alignment for headlight lamps from 3/64 in. to the decimal form. It was explained that the lamp manufacturers have all made this change. It was finally decided that the conversion should be made to the nearest three-point decimal value, thus converting the 3/64-in. tolerance to 0.047 in. The advisability of placing dimensions on the corrugations of the glass bulbs, which has been included in the Subdivision's report, was considered carefully and the conclusion reached that the actual form of the corrugations is not important so long as they are sufficient to break up the filament image. It was also considered too difficult and impractical to include these detail dimensions in the standard. The note relating to the corrugations was, however, added in the text of the report as given below. The form of the Subdivision's report was also modified by reducing the number of illustrations of types of incandescent lamps and giving their dimensions in tabular form.

It was felt that a valuable supplement to the standard would be a table listing the several incandescent lamps in commercial use, giving their candlepower, voltage, maximum amperage, and other information. The dimensions of the standard lamps are included in the table given below, which is intended only for general information to serve as a guide to the vehicle designer in selecting lamps.

The Lighting Division, therefore, submits the following report with the recommendation that it be adopted in revision of the present S.A.E. Standard for the Focusing Type of Electric Incandescent Lamp.





ELECTRIC INCANDESCENT LAMPS

Dimension	G-6	G-8	S-8	G-10	S-11[a]
A	3/4	1	1	1 1/4	1 3/8
B	1 7/16	1 3/4	2	2 1/8	2 3/8
C[1]	3/4	7/8	1 1/8	1 1/8	1 1/4
D Max.	0.300	0.300	0.300	0.300	0.300

[a] The corrugations on electric incandescent lamps for headlight service shall be of sufficient depth to break up the filament image.

The spacing between the filaments of the depressed-beam or two-filament type of incandescent lamp shall be 0.140 in. ± 0.016 in.

[1] Light-center length and axial alignment tolerances for headlight lamps are ± 0.047 in.

ELECTRIC INCANDESCENT LAMP DATA

General Information

Candle-Power	Voltage	Maximum Amperage[2]	Bayonet-Base Contact[3]	Bulb Type	Bulb Diameter, In.	Maximum Over-All Length, In.	Light Center Length, In.	Filament Form[4]
2	3-4	1.00	S or D	G-6	3/4	1 7/16	3/4	C-2
3	6-8	0.75	S or D	G-6	3/4	1 7/16	3/4	C-2
3	12-16	0.50	S or D	G-6	3/4	1 7/16	3/4	C-2
3	18-24	0.25	S or D	G-6	3/4	1 7/16	3/4	C-2
6	6-8	1.25	S or D	G-8	1	1 3/4	7/8	C-2
6	12-16	0.75	S or D	G-8	1	1 3/4	7/8	C-2
6	40-44	0.25	D	G-10	1 1/4	2 1/8	1 1/8	C-5
15	6-8	2.00	S or D	S-8	1	2	1 1/8	C-2
15	12-16	1.25	S or D	S-8	1	2	1 1/8	C-2
21 }	6-8	{3.00}	D	S-11	1 3/8	2 3/8	{1 1/4	C-2
21 }		{3.00}					{1 1/4	C-2
21	6-8	3.00	S or D	S-11	1 3/8	2 3/8	1 1/4	C-2
21	12-16	1.50	S or D	S-11	1 3/8	2 3/8	1 1/4	C-2
21	40-44	0.50	D	S-11	1 3/8	2 3/8	1 1/4	C-13
21 }	6-8	{3.00}	D	{S-11	1 3/8	2 3/8	1 1/4	C-2
2 }		{0.75}		{S-11	1 3/8	2 3/8		C-12
27	18-24	1.25	D	S-11	1 3/8	2 3/8	1 1/4	C-2
32	6-8	4.00	S or D	S-11	1 3/8	2 3/8	1 1/4	C-2
32	12-16	2.00	S or D	S-11	1 3/8	2 3/8	1 1/4	C-2

[2] Improvements in lamp design and manufacture from time to time make possible changes in ampere ratings. The figures given are therefore maximum and are for use in calculating wire sizes and battery capacities. For test purposes the exact amperage should be obtained from the lamp manufacturer.

[3] S—Single-contact. D—Double-contact.

[4] C indicates a coiled-wire filament.

MIDSHIP SHAFT MOUNTINGS

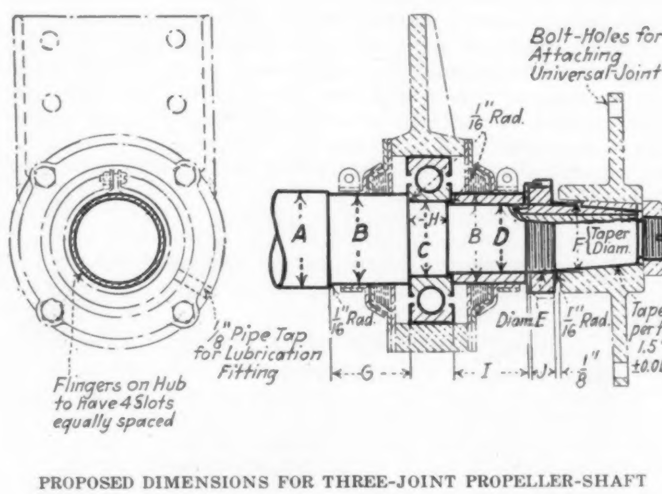
Motor Truck Division Recommends New Three-Joint Propeller-Shaft Specification

The proposed specification for propeller-shaft midship mountings was approved at a meeting of the Subdivision on Three-Joint Propeller-Shafts, held in Boston on Nov. 17, 1926, after consideration was given to numerous comments and suggestions received with regard to the previous report that was printed on p. 450 of the November, 1926, issue of THE JOURNAL. While the proposed S.A.E. Recommended Practice shows the bearing housing, this is not a part of the specification and is shown as a suggested design only. Some criticism of the proposal was made on the basis that it does not permit the use of any but the self-aligning type of bearing, but this was met with statements of other bearing manufacturers that the non-self-aligning types of bearing are used as well, especially in the case of bearings with a little looseness in the raceways. The question regarding heavy-duty installations was also raised, but the Subdivision feels that the recommendation will meet all usual practice, and that, as particularly heavy installations should be considered special, the shaft dimensions for heavier bearings should be agreed upon between the manufacturer and the purchaser.

Although the Subdivision report was not considered by the Motor Truck Division at a meeting, it has been submitted to letter-ballot by the Division for approval and adoption as S.A.E. Recommended Practice. In submitting the report to the Division for letter-ballot, the Standards Department sent the members information regarding the consideration that has been given the report by the Subdivision and vehicle manufacturers and references to discussion of the report in previous issues of THE JOURNAL.

THREE-JOINT PROPELLER-SHAFT MIDSHIP MOUNTING

The type of housing shown is recommended as a satisfactory design, but is not a part of the standard. No slip-joint in the forward section of the shaft is required, as clearance is provided in the housing for the bearing and shaft to slide longitudinally. Where a slip-joint is used, the housing should be designed according to requirements. The shafts and bearings recommended are considered adequate for all usual installations. Where heavier duty may require larger bearings, the shaft dimensions should be determined by the manufacturer and the purchaser.



PROPOSED DIMENSIONS FOR THREE-JOINT PROPELLER-SHAFT MIDSHIP MOUNTING

Tube Diameter, *A*, in.	2	2	2 1/4	2 1/2	3
B, + 0.005, — 0 in.	1.6870	1.8750	2.1250	2.3750	2.5620
C, max.	1.3780	1.5748	1.7717	1.9685	2.1654
C, min.	1.3772	1.5740	1.7709	1.9677	2.1646
D, + 0, — 0.003 in.	1.3740	1.4990	1.6240	1.8740	2.1240
E	1 3/8-18	1 1/2-18	1 5/8-16	1 7/8-16	2 1/8-16
S.A.E. Taper Shaft End, *F*	1 1/4	1 3/8	1 1/2	1 3/4	2
G, in.	1 5/8	1 3/4	1 7/8	2	2 1/8
H, in.	43/64	27/32	53/64	1	1 5/64
I, in.	1 9/16	1 11/16	1 13/16	1 15/16	2 1/16
J, in.	7/16	7/16	9/16	9/16	11/16
Bearing No.	307	308	309	310	311

Note.—The oil-retainer rings should be adjusted for minimum clearance after the housing and shaft are assembled in the chassis.

MOTOR TRUCK DIVISION PERSONNEL

The members of the Motor Truck Division are as follows:

B. B. Bachman, *Chairman*	Autocar Co.
A. W. S. Herrington, *Vice-Chairman*	City of Washington
N. G. Anderson	International Harvester Co.
W. J. Baumgartner	Garford Motor Truck Co.
M. C. Horine	International Motor Co.
D. F. Myers	Service Motors, Inc.
A. J. Scaife	White Motor Co.
F. J. Scarr	Pennsylvania Railroad
E. M. Sternberg	Sterling Motor Truck Co.
C. B. Veal	Manly & Veal
F. A. Whitten	American Car & Foundry Motors Co.
Ernest Wooler	Timken Roller Bearing Co.

FLEXIBLE-DISC TOLERANCES REVISED

Parts and Fittings Division Reverses Tolerances on Bolt-Hole Diameter

The Parts and Fittings Division at a meeting held in the General Motors Building, Detroit, on **Dec. 17** took under consideration the existing tolerances on flexible discs and the determination of a tolerance for thickness variation, these items having been the subject of a questionnaire recently sent to manufacturers and users of flexible discs.

As a result it has been recommended to change the bolt-hole dimension from basic diameter plus 0.010 in. to basic diameter minus 0.010 in. thereby reversing the present tolerances. It was further decided to retain the present tolerances on disc thickness.

The replies to the questionnaire indicate a need for standard tolerances on thickness variation. The difficulty of determining tolerances practicable for both users and manufacturers make it desirable to have the subject thoroughly gone into by a Subdivision. C. W. Spicer, of the Spicer Mfg. Corporation, and Walter C. Keys, of the United States Rubber Co., were appointed as this Subdivision and will present a report at the next meeting of the Division.

PARTS AND FITTINGS DIVISION PERSONNEL

The personnel of the Parts and Fittings Division is as follows:

H. S. Jandus, *Chairman*	C. G. Spring & Bumper Co.
A. Boor, *Vice-Chairman*	Willys-Overland Co.
Joseph Berge	Joseph Berge Co.
R. V. Hutchinson	Olds Motor Works
W. C. Keys	Detroit
G. R. Oliver	General Motors Truck Co.
W. J. Outcalt	General Motors Corporation
C. W. Spicer	Spicer Mfg. Corporation
F. C. Stanley	Raybestos Co.
H. L. Walker	Chandler Motor Car Co.
E. W. Weaver	Geo. T. Trundle, Jr., Engineering Co.
F. G. Whittington	Stewart-Warner Speedometer Corporation

TABLE OF ROD-END PINS REVISED

Under-Head to Hole-Center Dimension To Be Expressed in Fractions

The present table showing rod-end pin and cotter-pin dimensions gives dimensions to four decimal places for the distance from under the head to the center of the hole in rod-end pins.

The members of the Parts and Fittings Division at a meeting on Dec. 17 agreed that the showing of this dimension in four decimal places without any tolerances indicated a degree of accuracy which was not required. The existing dimensions are decimal equivalents of fractions and it was decided to recommend that the table be revised to give this dimension in fractions rather than as now designated.

NEW STANDARD FOR RIVETS PROPOSED

Parts and Fittings Division Reports Specifications of Sectional Committee

The Society, as one of the sponsors for the Sectional Committee on Bolt, Nut and Rivet Proportions that was organized under the procedure of the American Engineering Standards Committee, has received the reports of Subcommittee No. 1 on Small Rivets, 7/16-in. nominal diameter and under, and on Tinners', Coopers' and Belt Rivets, for approval. These reports were formally submitted to the Society by the Chairman and Secretary of the Sectional Committee on Nov. 11, 1926.

The personnel of the Subcommittee that prepared the reports is as follows:

H. N. Wallin, *Chairman*	Bureau of Construction and Repair, Navy Department
E. J. Edwards	American Locomotive Co.
Hugo P. Frear	Bethlehem Shipbuilding Corporation
Frank O. Kichline	Bethlehem Steel Co.
Richard W. Knight	McClintic-Marshall Production Co.
Philip G. Lang, Jr.	Baltimore & Ohio Railroad
William A. McKinley	Detroit Pressed Steel Co.
H. C. Weidner	Townsend Co.
Victor R. Willoughby	American Car & Foundry Co.
O. R. Wilson	Champion Rivet Co.

The reports state that all of the types of small rivet other than those included in the report on Small Rivets coming within the limits of diameters given in the report, are considered special. In the case of Tinners', Coopers' and Belt Rivets, the types included in the report are considered standard for all of these classes of rivet; all others have been considered special.

At the meeting of the Screw-Threads Division in Detroit on Dec. 16, it was felt that the report on Rivets should be referred to the Parts and Fittings Division for approval rather than for such action to be taken by the Screw-Threads Division. The reports were accordingly discussed at the meeting of the Parts and Fittings Division held in Detroit on Dec. 17. That part of the report relating to Small Rivets, 7/16 in. nominal diameter and under, was approved for adoption as tentative American Standard under procedure of the American Engineering Standards Committee, with the proviso that the term "shank" throughout the report be changed to "body" to conform with the rest of the reports of the Sectional Committee, and also that such approval be subject to confirmation by letter ballot of the Division members. It was also decided to refer the report to the Rivets Subdivision for recommendations as to its adoption, in part or in whole, as S.A.E. Standard.

The Parts and Fittings Division was assigned the subject of rivets some time ago and a Subdivision was thereupon appointed, consisting of A. Boor, of the Willys-Overland Co., chairman; W. J. Outcalt, of the General Motors Corporation; and Dr. F. C. Stanley, of the Raybestos Co. The scope of this assignment was, however, limited to the tubular and bifurcated types of rivet ordinarily used for such applications as brake-linings and clutch facings.

SMALL RIVETS

Tabular Diameters.—The diameters of rivets in fractions of an inch as given for the respective types of rivet in Tables 1, 2, 3, 4, and 5, shall be considered standard. Other values for diameters of rivets, as in Birmingham Wire Gage numbers, may be used in catalogs in conjunction with the standard diameters; it being recommended, however, that the data be in such form as will make clear which diameters are standard and which are not standard.

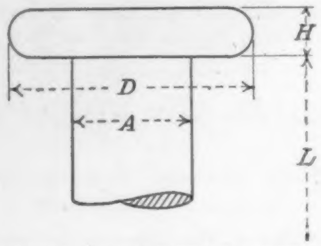


TABLE 1—FLAT-HEAD RIVETS

A			D	H	L
Diameter of Body			Diameter of Head	Height of Head	Length under Head
Nominal	Maximum	Minimum			
3/32—0.094	0.096	0.090	0.190	0.032	Ordered length under head to end.
1/8—0.125	0.127	0.121	0.250	0.042	
5/32—0.156	0.158	0.152	0.312	0.052	
3/16—0.188	0.191	0.182	0.374	0.062	
7/32—0.219	0.222	0.213	0.440	0.073	
1/4—0.250	0.253	0.244	0.500	0.083	
9/32—0.281	0.285	0.273	0.562	0.094	
5/16—0.313	0.317	0.305	0.624	0.104	
11/32—0.344	0.348	0.336	0.686	0.114	
3/8—0.375	0.380	0.365	0.750	0.125	
7/16—0.438	0.443	0.428	0.874	0.146	

Approximate Proportions. $D=2.00 \times A$
$H=0.33 \times A$

Proportions and Dimensions.—The proportions and dimensions indicated in the tables for the heads of the respective rivets shall be standard; other proportions or dimensions are to be considered special. Where non-standard diameters are given for rivets, the proportions of heads will follow the diagrammatic proportions given in the respective diagrams, in terms of the diameter of body.

Fillets.—Rivets, other than of the countersunk type, shall be acceptable with fillets under the head up to 1/32 in. radii.

Tolerances.—The tolerances on the nominal diameters of the bodies shall be those given in the following table:

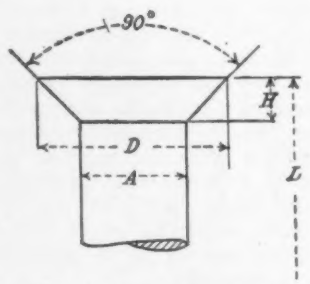


TABLE 2—COUNTERSUNK-HEAD RIVETS

A			D	H		L
Diameter of Body			Diameter of Head	Depth of Head	Included Angle, Deg.	Length over All
Nominal	Maximum	Minimum				
3/32—0.094	0.096	0.090	0.176	0.040	90	Ordered length.
1/8—0.125	0.127	0.121	0.231	0.053	90	
5/32—0.156	0.158	0.152	0.289	0.066	90	
3/16—0.188	0.191	0.182	0.346	0.079	90	
7/32—0.219	0.222	0.213	0.407	0.094	90	
1/4—0.250	0.253	0.244	0.463	0.106	90	
9/32—0.281	0.285	0.273	0.520	0.119	90	
5/16—0.313	0.317	0.305	0.577	0.133	90	
11/32—0.344	0.348	0.336	0.635	0.146	90	
3/8—0.375	0.380	0.365	0.694	0.159	90	
7/16—0.438	0.443	0.428	0.808	0.186	90	

Approximate Proportions. $D=1.850 \times A$
$H=0.425 \times A$

Diameters of Body, Inclusive	Tolerances Plus	Minus
3/32 — 5/32	0.002	0.004
3/16 — 1/4	0.003	0.006
9/32 — 11/32	0.004	0.008
3/8 — 7/16	0.005	0.010

No standard tolerances for the dimensions of the heads are contemplated. For work where restrictions as to head tolerances are necessary, these shall be considered special.

Rivet Material: Composition and Physical Properties.—(*a*) The steel from which the rivets are manufactured shall be made by the open-hearth process and conform to the following:

Manganese, per cent................0.30 to 0.50
Phosphorus, maximum, per cent........... 0.04
Sulphur, maximum, per cent.............. 0.05

(*b*) The material, when tested, shall conform to the following:

Tensile-strength, lb. per sq. in., 45,000 to 55,000
Yield point, lb. per sq. in., 0.5 tensile-strength
Elongation in 8 in., minimum per cent =
1,500,000/Tensile-strength
The elongation need not, however, exceed 30 per cent.

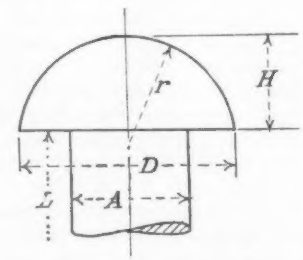


TABLE 3—BUTTON-HEAD RIVETS

A			D	H	r	L
Diameter of Body			Diameter of Head	Height of Head	Radius of Top	Length under Head
Nominal	Maximum	Minimum				
3/32—0.094	0.096	0.090	0.166	0.071	0.084	Ordered length under head to end.
1/8—0.125	0.127	0.121	0.219	0.094	0.111	
5/32—0.156	0.158	0.152	0.273	0.117	0.138	
3/16—0.188	0.191	0.182	0.327	0.140	0.166	
7/32—0.219	0.222	0.213	0.385	0.165	0.195	
1/4—0.250	0.253	0.244	0.438	0.188	0.221	
9/32—0.281	0.285	0.273	0.492	0.211	0.249	
5/16—0.313	0.317	0.305	0.546	0.234	0.276	
11/32—0.344	0.348	0.336	0.600	0.257	0.304	
3/8—0.375	0.380	0.365	0.656	0.281	0.332	
7/16—0.438	0.443	0.428	0.765	0.328	0.387	

Approximate Proportions. $D=1.750 \times A$
$H=0.750 \times A$
$r=0.885 \times A$

The preceding requirements are not applicable to tests on finished rivets. They are for general information only in regard to the production of suitable rivet material.

Physical Tests.—Rivets selected at random from each size shall comply with the following:

(*a*) *Cold Test for Ductility*

One-half of the rivets selected for each test shall be flattened to one-fourth of their original diameter and then bent through 180 deg. flat on themselves and shall show no signs of cracks, flaws or any other defects.

(*b*) *Hot Test for Ductility*

The remaining rivets shall be heated to a red heat and flattened to one-fourth of their original diameter, then re-heated and bent through 180

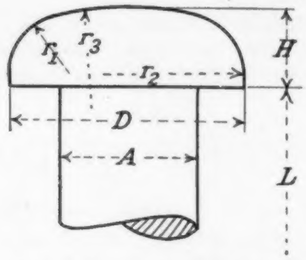


TABLE 4—PAN-HEAD RIVETS

A			D	H	r			L
Diameter of Body			Diameter of Head	Height of Head	Radii of Head			Length under Head
Nominal	Maximum	Minimum			r_1	r_2	r_3	
3/32—0.094	0.096	0.090	0.163	0.054	0.030	0.080	0.326	Ordered length under head to end.
1/8 —0.125	0.127	0.121	0.215	0.072	0.039	0.106	0.429	
5/32—0.156	0.158	0.152	0.268	0.089	0.049	0.133	0.535	
3/16—0.188	0.191	0.182	0.321	0.107	0.059	0.159	0.641	
7/32—0.219	0.222	0.213	0.378	0.126	0.069	0.186	0.754	
1/4 —0.250	0.253	0.244	0.429	0.143	0.079	0.213	0.858	
9/32—0.281	0.285	0.273	0.482	0.161	0.088	0.239	0.963	
5/16—0.313	0.317	0.305	0.535	0.178	0.098	0.266	1.070	
11/32—0.344	0.348	0.336	0.589	0.196	0.108	0.292	1.176	
3/8 —0.375	0.380	0.365	0.644	0.215	0.118	0.319	1.286	
7/16—0.438	0.443	0.428	0.750	0.250	0.137	0.372	1.500	

Approximate Proportions. $D = 1.720 \times A$
$H = 0.570 \times A$
$r_1 = 0.314 \times A$
$r_2 = 0.850 \times A$
$r_3 = 3.430 \times A$

deg. flat on themselves and shall show no signs of cracks, flaws or any other defects.

(*c*) *Hardness Test*

The hardness of rivets shall register between No. 20 and No. 26, inclusive, when tested by the Shore scleroscope. Any other degrees of hardness shall be considered special.

Finish.—The finished rivets shall be free from injurious defects.

Tinners', Coopers' and Belt Rivets

That part of the report relating to Tinners', Coopers' and Belt Rivets was not approved because the rivet body diameters, which are the most important dimensions, are given to two decimal places only whereas the tolerances on these diameters and the diameter and height of heads are given to three decimal places. It was also pointed out that the body diameter of some sizes of the rivets does not conform with the dimensions used by the rivet manufacturers. Criticisms were also made of the lack of clarity in the wording in some of the footnotes and it was felt that the report should be more thoroughly reviewed and revised before it is approved. It was agreed that the Chairman of the Division would submit by letter the reasons for not approving this report at this time so that they may be transmitted to the Sectional Committee. The report on Tinners', Coopers' and Belt Rivets is included in this issue of The Journal to give the members an opportunity to submit to the Standards Department of the Society any further criticisms of this report so that they may also be transmitted to the Sectional Committee.

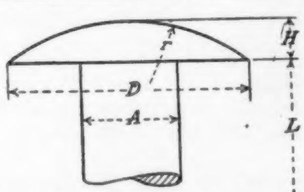


TABLE 5—TRUSS-HEAD RIVETS

A			D	H	r	L
Diameter of Body			Diameter of Head	Height of Head	Radius of Top	Length under Head
Nominal	Maximum	Minimum				
3/32—0.094	0.096	0.090	0.238	0.032	0.239	Ordered length under head to end.
1/8 —0.125	0.127	0.121	0.313	0.042	0.314	
5/32—0.156	0.158	0.152	0.390	0.052	0.392	
3/16—0.188	0.191	0.182	0.468	0.062	0.470	
7/32—0.219	0.222	0.213	0.550	0.073	0.555	
1/4 —0.250	0.253	0.244	0.625	0.083	0.628	
9/32—0.281	0.285	0.273	0.703	0.094	0.706	
5/16—0.313	0.317	0.305	0.780	0.104	0.784	
11/32—0.344	0.348	0.336	0.858	0.114	0.862	
3/8 —0.375	0.380	0.365	0.938	0.125	0.942	
7/16—0.438	0.443	0.428	1.093	0.146	1.098	

Approximate Proportions. $D = 2.500 \times A$
$H = 0.330 \times A$
$r = 2.512 \times A$

Tabular Sizes—The sizes of rivets as given for the respective types of rivet in Tables 1, 2 and 3 shall be considered standard. Other values for sizes of rivets may be used in catalogs in conjunction with the standard sizes; it being recommended, however, that the data be in such form as will make clear which are standard and which are not standard.

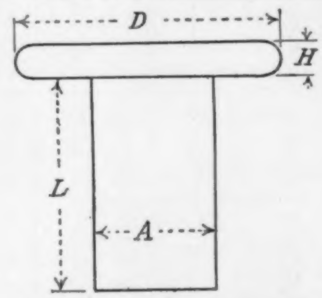


TABLE 1—TINNERS' RIVETS

Size No.	A	D	H	L
	Diameter of Body	Diameter of Head	Height of Head	Length under Head
8 oz.	0.09	0.203	0.027	0.15
12 oz.	0.10	0.225	0.030	0.18
1 lb.	0.11	0.248	0.033	0.20
1½ lb.	0.13	0.293	0.039	0.23
2 lb.	0.14	0.315	0.042	0.26
2½ lb.	0.15	0.338	0.045	0.28
3 lb.	0.16	0.360	0.048	0.31
4 lb.	0.17	0.383	0.051	0.34
6 lb.	0.20	0.450	0.060	0.39
8 lb.	0.22	0.495	0.066	0.43
10 lb.	0.24	0.540	0.072	0.46
12 lb.	0.26	0.575	0.078	0.50
14 lb.	0.28	0.630	0.084	0.51
16 lb.	0.30	0.675	0.090	0.53

Approximate Proportions. $D = 2.25 \times A$
$H = 0.30 \times A$

Proportions—The proportions indicated below, in Tables 1, 2 and 3, for the heads and points of the respective sizes of rivets shall be standard; other proportions are to be considered special. In non-standard sizes of rivet, the heads and points shall be of the same proportions; the lengths of rivets covered by Tables 1 and 2 being determined from the table by interpolation.

Tolerances—The tolerances on the various sizes of coopers' and tinners' rivets shall be those given in the following table:

Size, Inclusive	Tolerances Plus	Tolerances Minus
8 oz.— 2½ lb.	0.002	0.004
3 lb.—10 lb.	0.003	0.006
12 lb.—16 lb.	0.004	0.008

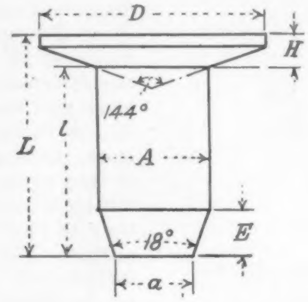


TABLE 2—COOPERS' RIVETS

Size No.	A	D		H	l	L	E	a
	Diameter of Body	Diameter of Head	Included Angle, Deg.	Height of Head	Length under Head	Over-All Length	Length of Chamfer	Diameter of Tip
1 lb.	0.11	0.248	144	0.033	0.18	0.213	0.044	0.099
1½ lb.	0.13	0.293	144	0.039	0.22	0.259	0.052	0.117
2 lb.	0.14	0.315	144	0.042	0.25	0.292	0.056	0.126
2½ lb.	0.15	0.338	144	0.045	0.28	0.325	0.060	0.135
3 lb.	0.16	0.360	144	0.048	0.31	0.358	0.064	0.144
4 lb.	0.17	0.383	144	0.051	0.34	0.391	0.068	0.153
6 lb.	0.20	0.450	144	0.060	0.40	0.460	0.080	0.180
8 lb.	0.23	0.518	144	0.069	0.50	0.569	0.092	0.207
10 lb.	0.25	0.563	144	0.075	0.53	0.605	0.100	0.225
12 lb.	0.26	0.585	144	0.078	0.53	0.608	0.104	0.234
14 lb.	0.27	0.608	144	0.081	0.56	0.641	0.108	0.243
16 lb.	0.28	0.630	144	0.084	0.59	0.674	0.112	0.252

Approximate Proportions. $a = 0.90 \text{x} A$
$D = 2.25 \text{x} A$
$E = 0.40 \text{x} A$
$H = 0.30 \text{x} A$

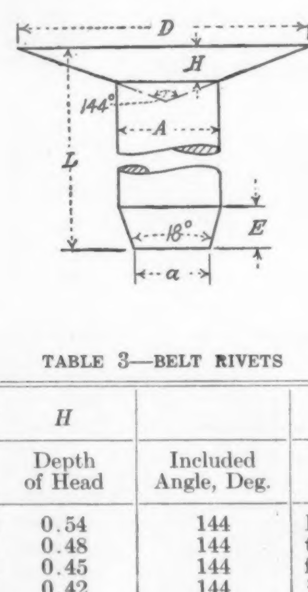


TABLE 3—BELT RIVETS

Size No.	A	D	H		L	E	a
	Diameter of Body	Diameter of Head	Depth of Head	Included Angle, Deg.	Length over All	Length of Chamfer	Diameter of Tip
7	0.18	0.504	0.54	144	Lengths	0.072	0.162
8	0.16	0.448	0.48	144	to be	0.064	0.144
9	0.15	0.420	0.45	144	from	0.060	0.135
10	0.14	0.392	0.42	144	⅜	0.056	0.125
					to		
11	0.12	0.336	0.36	144	¾ in.	0.048	0.108
12	0.10	0.280	0.30	144	by	0.040	0.090
13	0.09	0.252	0.27	144	⅛-in. increments	0.036	0.081

Approximate Proportions. $a = 0.9 \text{x} A$
$D = 2.8 \text{x} A$
$E = 0.4 \text{x} A$
$H = 0.3 \text{x} A$

Tolerances. The tolerances on the nominal diameter of Belt Rivets shall be plus 0.002 in. and minus 0.004 in.

No standard tolerances for the dimensions of the heads are contemplated. For work where restrictions as to head tolerances are necessary, these shall be considered special.

HIGH NUTS TO BE CHAMFERED

Screw-Threads Division Recommends 40-Deg. Finish in Present Standard

When the present S.A.E. Standard for High Nuts, p. C3f of the S.A.E. HANDBOOK, was adopted by the Society, the chamfer was intentionally omitted, as standard practice for thick nuts was expected to be followed. Since then, however, the question of having the standard specific in this regard has been raised and the Screw-Threads Division, the personnel of which is printed below, feels that the maximum chamfer should be included to prevent cutting down the sides of the nuts too deeply and defeating their purpose which is to provide a long wrench hold. Some difference in practice as to the amount of chamfer, varying from 30 to 45 deg. with the top face of the nut, seems to exist, and the Division, after considering the suggestions offered, has voted to recommend that high nuts shall be chamfered to the maximum angle of 40 deg. with the top face of the nut without specifying limits. This recommendation will prevent chamfering the nuts too deeply and permit of a shallower chamfer when desired.

SCREW-THREADS DIVISION PERSONNEL

E. H. Ehrman, *Chairman*	Standard Screw Co.
G. S. Case, *Vice-Chairman*	Lamson & Sessions Co.
A. Boor	Willys-Overland Co.
E. J. Bryant	Greenfield Tap & Die Corporation
Earle Buckingham	Massachusetts Institute of Technology
Ellwood Burdsall	Russell, Burdsall & Ward Bolt & Nut Co.
L. D. Burlingame	Brown & Sharpe Mfg. Co.
R. M. Heames	Victor-Peninsular Co.
K. L. Herrmann	Studebaker Corporation of America
D. W. Ovaitt	Buick Motor Co.
O. B. Zimmerman	International Harvester Co.

CAP-SCREW, BOLT AND NUT REVISION

Short Screw-Thread Length Specification Approved by Screw-Threads Division

The Screw-Threads Division at a meeting held in the General Motors Building, Detroit, on Dec. 16 discussed the revision of the specification covering cap screw-thread lengths.

The Division recommends that the tables covering short lengths be eliminated from the standards and has approved the following specification for short screw-thread lengths:

> Screws too short to take the formula length of thread may be threaded as close to the head as practicable.
>
> It is further recommended that the words "usable" and "visible" in the specifications for thread length of screws taking the formula length of thread be deleted from the specification.

The several tables on cap screws and nuts will be revised by the addition of a column showing the thread pitches and a general rearrangement made to facilitate the usage of these standards.

PLOW-BOLT STANDARD SUBMITTED

Sectional Committee Report Submitted for Approval by the Society as a Sponsor

Plow bolts, which are a distinct class of bolt used in the construction of agricultural implements, are one classification for these products that has been under consideration by the Sectional Committee on Bolt, Nut and Rivet Proportions that was organized under the procedure of the American Engineering Standards Committee and sponsored jointly by the Society and the American Society of Mechanical Engineers. After reviewing data on the large variety of plow bolts in use, the Subcommittee that was appointed learned that a similar program of standardization was being carried on by a Committee appointed by the Executive Committee of the National Association of Farm Equipment Manufacturers. To coordinate the work of the two Committees was deemed advisable and accordingly the Subcommittee of the Sectional Committee held its work in abeyance until the Committee of the National Association of Farm Equipment Manufacturers had developed a report, as that Committee was considered as representing the largest single group of farm equipment manufacturers in this Country. The National Association of Farm Equipment Manufacturers Committee eventually reported seven types of plow bolt representing approximately 182 varieties. From these, four types representing 42 varieties were finally selected as being sufficient to meet the demands of the agricultural industry. These were discussed at a conference called by the Division of Simplified Practice, Department of Commerce, at the City of Washington, in February, 1924. Shortly thereafter the Subcommittee of the Sectional Committee was reorganized with the following personnel to include representation of the National Association of Farm Equipment Manufacturers:

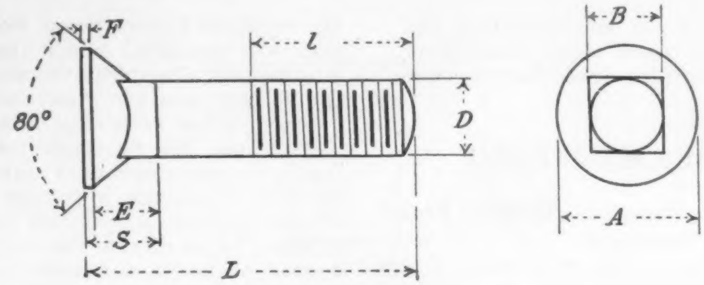
E. P. Stahl, *Chairman and Secretary*	Hyatt Roller Bearing Co.
Elwood Burdsall	Russell, Burdsall & Ward Bolt & Nut Co.
George S. Case	Lamson & Sessions Co.
Charles B. Segner	Domestic Engine & Pump Co.
Oliver B. Zimmerman	International Harvester Co.

The report of this Subcommittee has been approved by the Sectional Committee and submitted to the Society as one of the sponsors, for approval as tentative American Standard. The Screw-Threads Division to which the report has been assigned was sent advance copies as approved by the Sectional Committee and has considered the report at its meeting in Detroit on Dec. 16, 1926. The dimensions for the regular and repair bolt were approved for adoption as tentative American Standard under American Engineering Standards Committee procedure with the proviso that this approval be confirmed by letter ballot of the Division members. The Division disapproved including the body and thread lengths for the same reasons as referred to in the report on Round Unslotted-Head Bolts printed on p. 27 of this issue of THE JOURNAL. It was also suggested that in printing the tables of the bolt dimensions in the American Standard report, the old plow bolt numbers 3, 4, 6, and 7 be taken out of the main captions and included in notes to the effect that these are the numbers by which these bolts were known previous to their standardization in this report. It was felt that if numbers are given to the standard bolts, they should be a new consecutive series. The Division therefore submits it at this time only for approval by the Society as a sponsor under the procedure of the American Engineering Standards Committee. The report is printed in this issue of THE JOURNAL for the information and review of the members of the Society, prior to its being passed upon by the Standards Committee during the Annual Meeting of the Society in Detroit, Jan. 25 to 28.

PLOW BOLTS

No. 3. Round Countersunk Short-Square Type.—This is generally used in steel parts where countersinking the holes is necessary. It is also used to some extent where the hole can be either dry-sand or green-sand cored. With an included angle of the head of 80 deg., the hole can be countersunk conveniently leaving a sufficiently strong head without too much top surface which might be objectionable in soft center moldboards where scouring is essential.

No. 4. Square-Head Countersunk Type.—This bolt is used only in cast-iron or malleable parts, where the



ROUND COUNTERSUNK, SHORT SQUARE PLOW BOLT

Regular Bolt

D		A		B			S	F	E
Nominal Diameter of Bolt	Threads per Inch	Head Diameter		Square Neck[1]		Included Angle of Head, Deg.	Minimum	Feed Thickness Maximum	Minimum
		Minimum	Tolerance +	Maximum	Tolerance −				
5/16	18	0.562 (9/16)	0.015	0.312	0.012	80	0.243	0.025	0.218
3/8	16	0.656 (21/32)	0.015	0.375	0.012	80	0.281	0.031	0.250
7/16	14	0.765 (49/64)	0.015	0.437	0.012	80	0.328	0.036	0.292
1/2	13	0.875 (7/8)	0.015	0.500	0.015	80	0.375	0.042	0.333
9/16	12	0.968 (31/32)	0.031	0.562	0.015	80	0.416	0.045	0.371
5/8	11	1.062 (1 1/16)	0.031	0.625	0.015	80	0.456	0.050	0.406
3/4	10	1.219 (1 7/32)	0.031	0.750	0.015	80	0.491	0.050	0.441

Repair Bolt

D		a		b			s	f	e
Nominal Diameter of Bolt	Threads per Inch	Head Diameter		Square Neck		Included Angle of Head, Deg.	Minimum	Feed Thickness Maximum	Minimum
		Minimum	Tolerance +	Maximum	Tolerance −				
5/16	18	0.516 (33/64)	0.015	0.312	0.012	80	0.212	0.020	0.192
3/8	16	0.609 (39/64)	0.015	0.375	0.012	80	0.247	0.025	0.222
7/16	14	0.719 (23/32)	0.015	0.437	0.012	80	0.294	0.030	0.264
1/2	13	0.828 (53/64)	0.015	0.500	0.015	80	0.340	0.035	0.305
9/16	12	0.922 (59/64)	0.031	0.562	0.015	80	0.383	0.040	0.343
5/8	11	1.016 (1 1/64)	0.031	0.625	0.015	80	0.418	0.040	0.378
3/4	10	1.172 (1 11/64)	0.031	0.750	0.015	80	0.453	0.040	0.413

[1] These dimensions for cold upset wire

BOLT AND THREAD LENGTHS

Nominal Diameter of Bolt	5/16	3/8	7/16	1/2	9/16	5/8	3/4	Length Tolerance
Length of Bolt, L	Minimum Length of Thread, l							+ or −
3/4	T to H	T to H	T to H	....	....	....	....	1/32
7/8	T to H	T to H	T to H	T to H	....	....	....	1/32
1	T to H	T to H	T to H	T to H	....	....	....	1/32
1 1/8	11/16	3/4	5/8	T to H	....	....	....	1/32
1 1/4	3/4	3/4	5/8	T to H	T to H	T to H	....	1/32
1 3/8	7/8	3/4	5/8	3/4	T to H	T to H	....	1/32
1 1/2	1	7/8	7/8	7/8	3/4	7/8	....	1/32
1 3/4	7/8	7/8	7/8	7/8	7/8	1 1/8	....	1/16
2	7/8	7/8	7/8	7/8	1 1/8	1 1/8	....	1/16
2 1/4	....	7/8	1	1	1 1/8	1 1/4	1 1/4	1/16
2 1/2	....	1	1	1	1 1/4	1 1/4	1 1/4	1/16
2 3/4	....	1	1	1	1 1/4	1 1/4	1 1/4	1/16
3	....	1	1	1	1 1/4	1 1/4	1 1/4	1/16

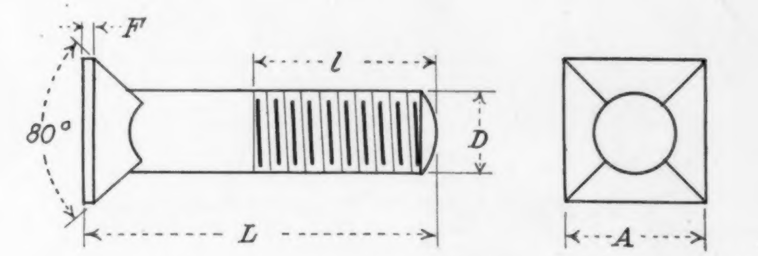
hole is green-sand cored. The head with the 80 deg. angle is rather flat and, therefore, is desirable for use in cast shares that are very often very thin. The 80-deg. angle was adopted for this bolt as a fair compromise and because it is now used by two of the largest plow manufacturers.

No. 6. Heavy Key Round-Head Countersunk Type.—This bolt is used in chilled and cast moldboards where the holes are always dry-sand colored. This bolt with its 40-deg. angle of head was adopted for the reason that three of the largest plow manufacturers now use it.

No. 7. Reverse-Key Round-Head Countersunk Type.—This bolt is used in steel parts. The key on this bolt being larger on the bottom makes it very desirable for parts of plows such as the moldboard for the reason that the head can wear down with the surface of the board without wearing away the entire key. The key does not extend outside the top portion of the bolt, and therefore, if the key on the bolt head does not fill the entire slot in the hole, this is not noticeable. The 60-deg. angle on this bolt makes it very desirable for plow-bottom work, as the head is not any larger than necessary. It is very important that the soft head of a bolt in a highly tempered and polished moldboard should be no larger than necessary on account of difficult conditions of scouring. The 60-deg. angle of head was adopted for this bolt because it is now used by several of the large plow manufacturers.

Screw Thread.—The threaded parts of these four standard plow bolts are to conform to the American National Standard Coarse-Thread Series—Free Fit (Class 2).

The tolerance on the included angle of the head shall

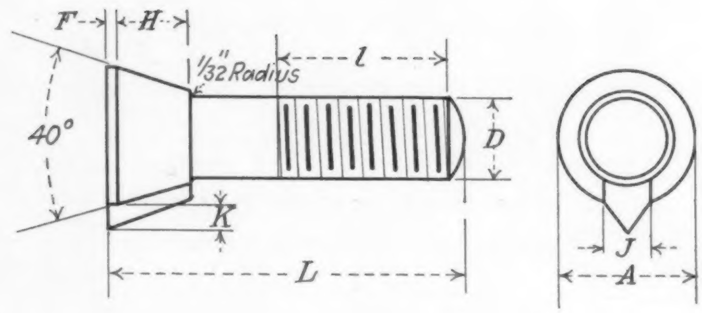


SQUARE-HEAD COUNTERSUNK PLOW BOLT

Regular Bolt							Repair Bolt						
D		A		F			D		a		f	
Nominal Diameter of Bolt	Threads per Inch	Head Diameter		Feed Thickness, Maximum	Included Angle of Head, Deg.		Nominal Diameter of Bolt	Threads per Inch	Head Diameter		Feed Thickness, Maximum	Included Angle of Head, Deg.
		Minimum	Tolerance +						Minimum	Tolerance +		
5/16	18	0.562 (9/16)	0.015 (1/64)	0.025	80		5/16	18	0.515 (33/64)	0.015 (1/64)	0.020	80
3/8	16	0.687 (11/16)	0.015 (1/64)	0.031	80		3/8	16	0.640 (41/64)	0.015 (1/64)	0.025	80
7/16	14	0.750 (3/4)	0.015 (1/64)	0.036	80		7/16	14	0.703 (45/64)	0.015 (1/64)	0.030	80
1/2	13	0.812 (13/16)	0.015 (1/64)	0.042	80		1/2	13	0.765 (49/64)	0.015 (1/64)	0.035	80
9/16	12	0.937 (15/16)	0.031 (1/32)	0.045	80		9/16	12	0.890 (57/64)	0.031 (1/32)	0.040	80
5/8	11	1.000 (1)	0.031 (1/32)	0.050	80		5/8	11	0.953 (61/64)	0.031 (1/32)	0.040	80
3/4	10	1.125 (1 1/8)	0.031 (1/32)	0.050	80		3/4	10	1.078 (1 5/64)	0.031 (1/32)	0.040	80
7/8	9	1.312 (1 5/16)	0.031 (1/32)	0.050	80		7/8	9	1.265 (1 17/64)	0.031 (1/32)	0.040	80
1	8	1.500 (1 1/2)	0.031 (1/32)	0.050	80		1	8	1.453 (1 29/64)	0.031 (1/32)	0.040	80

BOLT AND THREAD LENGTHS

Nominal Diameter	5/16	3/8	7/16	1/2	9/16	5/8	3/4	7/8	1	Length Tolerance
Length of Bolt, *L*	Minimum Length of Thread, *l*									+ or −
3/4	*T to H*	*T to H*	*T to H*	...	...	...	...	...	...	1/32
7/8	*T to H*	*T to H*	*T to H*	*T to H*	...	...	...	...	...	1/32
1	*T to H*	*T to H*	*T to H*	*T to H*	...	...	...	...	...	1/32
1 1/8	11/16	3/4	5/8	*T to H*	...	...	...	...	...	1/32
1 1/4	3/4	3/4	5/8	*T to H*	*T to H*	*T to H*	...	...	...	1/32
1 3/8	..	3/4	3/4	3/4	*T to H*	*T to H*	...	...	...	1/32
1 1/2	..	7/8	7/8	7/8	7/8	7/8	...	...	...	1/32
1 3/4	..	7/8	7/8	7/8	7/8	1	...	...	...	1/16
2	..	7/8	7/8	7/8	1 1/8	1 1/8	1 1/8	...	...	1/16
2 1/4	..	7/8	1	1	1 1/8	1 1/4	1 1/4	1 1/4	1 1/4	1/16
2 1/2	..	1	1	1	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1/16
2 3/4	..	1	1	1	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1/16
3	..	1	1	1	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1/16

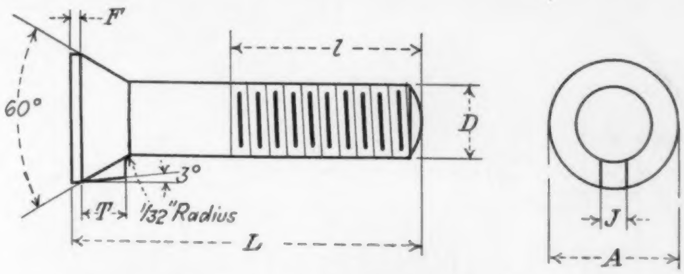


HEAVY KEY, ROUND HEAD, COUNTERSUNK PLOW BOLT

Regular Bolt									*Repair Bolt*		
D		*A*		*F*		*H*	*J*	*K*	*a*	*f*	*h*
Nominal Diameter	Threads per Inch	Head Diameter Minimum	Head Diameter Tolerance +	Feed Thickness Maximum	Included Angle of Head, Deg.	Height of Head Maximum	Key Width Maximum	Key Projection Maximum	Head Diameter Maximum	Feed Thickness Maximum	Height of Head Maximum
3/8	16	0.725	0.015	0.050	40	0.310	0.156	0.100	0.708	0.025	0.286
7/16	14	0.796	0.015	0.050	40	0.367	0.172	0.100	0.770	0.030	0.339
1/2	13	0.867	0.015	0.050	40	0.430	0.188	0.100	0.836	0.035	0.402

Bolt and Thread Lengths

Nominal Diameter of Bolt	3/8	7/16	1/2	Length Tolerance
Length of Bolt, L	Minimum Length of Thread, *l*			+ or −
3/4	*T to H*	*T to H*	*T to H*	1/32
7/8	*T to H*	*T to H*	*T to H*	1/32
1	*T to H*	*T to H*	*T to H*	1/32
1 1/8	*T to H*	*T to H*	*T to H*	1/32
1 1/4	*T to H*	*T to H*	*T to H*	1/32
1 3/8	*T to H*	*T to H*	*T to H*	1/32
1 1/2	15/16	7/8	7/8	1/32
1 3/4	7/8	7/8	7/8	1/16
2	7/8	7/8	7/8	1/16
2 1/4	7/8	1	1	1/16
2 1/2	7/8	1	1	1/16
2 3/4	1	1	1	1/16
3	1	1	1	1/16



REVERSE KEY, ROUND HEAD, COUNTERSUNK PLOW BOLT

Regular Bolt

D	Threads per Inch	A		F	Included Angle of Head, Deg.	J		T	
Nominal Diameter of Bolt		Head Diameter		Feed Thickness		Key Width		Key Length	
		Minimum	Tolerance +	Maximum		Maximum	Tolerance −	Minimum	Tolerance +
5/16	18	0.562 (9/16)	0.015	0.025	60	0.156	0.005	0.185	0.015
3/8	16	0.625 (5/8)	0.015	0.031	60	0.156	0.005	0.187	0.015
7/16	14	0.734 (47/64)	0.015	0.036	60	0.156	0.005	0.227	0.015
1/2	13	0.843 (27/32)	0.015	0.042	60	0.156	0.005	0.267	0.015
9/16	12	0.968 (31/32)	0.015	0.045	60	0.156	0.005	0.321	0.015
5/8	11	1.063 (1 1/16)	0.015	0.050	60	0.156	0.005	0.348	0.015
3/4	10	1.218 (1 7/32)	0.015	0.050	60	0.156	0.005	0.375	0.015

Repair Bolt

D	Threads per Inch	a		f	Included Angle of Head, Deg.	j		t	
Nominal Diameter of Bolt		Head Diameter		Feed Thickness		Key Width		Key Length	
		Minimum	Tolerance +	Maximum		Maximum	Tolerance −	Minimum	Tolerance +
5/16	18	0.531 (17/32)	0.015	0.020	60	0.156	0.005	0.158	0.015
3/8	16	0.593 (19/32)	0.015	0.025	60	0.156	0.005	0.158	0.015
7/16	14	0.703 (45/64)	0.015	0.030	60	0.156	0.005	0.199	0.015
1/2	13	0.812 (13/16)	0.015	0.035	60	0.156	0.005	0.239	0.015
9/16	12	0.937 (15/16)	0.015	0.040	60	0.156	0.005	0.293	0.015
5/8	11	1.031 (1 1/32)	0.015	0.040	60	0.156	0.005	0.321	0.015
3/4	10	1.187 (1 3/16)	0.015	0.040	60	0.156	0.005	0.347	0.015

BOLT AND THREAD LENGTHS

Nominal Diameter of Bolt	5/16	3/8	7/16	1/2	9/16	5/8	3/4	Length Tolerance
Length of Bolt, *L*	Minimum Length of Thread, *l*							+ or −
3/4	*T to H*	*T to H*	*T to H*	....	....	....	....	1/32
7/8	*T to H*	*T to H*	*T to H*	....	....	....	....	1/32
1	*T to H*	*T to H*	*T to H*	*T to H*	....	....	....	1/32
1 1/8	11/16	3/4	5/8	*T to H*	....	....	....	1/32
1 1/4	3/4	3/4	5/8	*T to H*	*T to H*	*T to H*	....	1/32
1 3/8	....	3/4	3/4	3/4	*T to H*	*T to H*	....	1/32
1 1/2	....	7/8	7/8	7/8	7/8	7/8	....	1/32
1 3/4	....	7/8	7/8	7/8	7/8	1	....	1/16
2	....	7/8	7/8	7/8	1 1/8	1 1/8	1 1/8	1/16
2 1/4	....	7/8	1	1	1 1/8	1 1/4	1 1/4	1/16
2 1/2	....	1	1	1	1 1/4	1 1/4	1 1/4	1/16
2 3/4	....	1	1	1	1 1/4	1 1/4	1 1/4	1/16
3	....	1	1	1	1 1/4	1 1/4	1 1/4	1/16

be plus 2 deg. The reference letters corresponding to the dimensions of the repair bolts are identical with those of the regular bolts but are printed in lower case type. To distinguish the repair bolts from regular bolts of the same type, a large *R* shall be stamped on the top of its head or this top surface shall be painted red. *T to H* means Threaded to Head.

ROUND UNSLOTTED-HEAD BOLTS

Proposed Tentative American Standard Submitted for Approval by Society

One of the projects of the Sectional Committee on Bolt, Nut and Rivet Proportions that was organized under the procedure of the American Engineering Standards Committee and sponsored by the American Society of Mechanical Engineers and the Society is the standardization of round unslotted-head bolts. The report of Subcommittee No. 5 that was organized to prepare the report on this class of bolt has submitted its report to the Sectional Committee, which approved the report and has submitted it to the Society for approval. The Subcommittee of the Sectional Committee submitting the report is comprised of the following:

William M. Horton, Jr., *Chairman*	Kirk-Latty Mfg. Co.
Ellwood Burdsall	Russell, Burdsall & Ward Bolt & Nut Co.
Merrill C. Horine	International Motor Co.
R. Plumb	Buffalo Bolt Co.
Charles B. Segner	Domestic Engine & Pump Co.
Edward P. Stahl	Hyatt Roller Bearing Co.
Oliver B. Zimmerman	International Harvester Co.

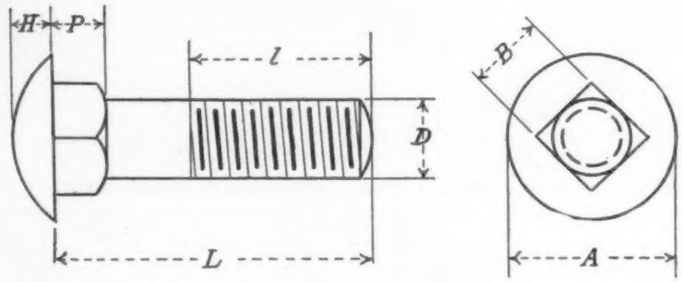


TABLE 1—SQUARE-NECK CARRIAGE-BOLT

Nominal Size	D		Threads per Inch	A		H		P		B	
	Major Diameter of Thread			Diameter of Head		Height of Head		Depth of Square		Width of Square	
	Maximum Basic	Tolerance −		Basic	Tolerance + or −	Basic	Tolerance + or −	Basic	Tolerance +	Basic	Tolerance −
No. 10	0.190	0.009	24	0.438 (7/16)	0.010	0.094	0.010	0.188	0.031	0.190	0.009
1/4	0.250	0.010	20	0.563 (9/16)	0.010	0.125	0.010	0.219	0.031	0.250	0.010
5/16	0.313	0.013	18	0.688 (11/16)	0.010	0.156	0.010	0.250	0.031	0.313	0.013
3/8	0.375	0.015	16	0.813 (13/16)	0.010	0.188	0.010	0.281	0.031	0.375	0.015
7/16	0.438	0.015	14	0.938 (15/16)	0.010	0.219	0.010	0.313	0.031	0.438	0.015
1/2	0.500	0.015	13	1.063 (1 1/16)	0.010	0.250	0.010	0.344	0.031	0.500	0.015
9/16	0.563	0.016	12	1.188 (1 3/16)	0.015	0.281	0.015	0.375	0.031	0.563	0.016
5/8	0.625	0.017	11	1.313 (1 5/16)	0.015	0.313	0.015	0.406	0.031	0.625	0.017
3/4	0.750	0.020	10	1.563 (1 9/16)	0.015	0.375	0.015	0.469	0.031	0.750	0.020

Nominal Size	No. 10	1/4	5/16	3/8	7/16	1/2	9/16	5/8	3/4	Length Tolerance + or −
Length of Bolt, *L*	Minimum Length of Thread, *l*, Including Point									
1/2	*T to S*	*T to S*	...	...	...	...	...	...	...	1/64
3/4	3/8	3/8	3/8	3/8	...	...	...	...	...	1/64
1	1/2	1/2	1/2	5/8	5/8	*T to S*	...	...	...	1/64
1 1/4	1/2	5/8	5/8	5/8	3/4	3/4	*T to S*	*T to S*	...	1/64
1 1/2	1/2	3/4	3/4	3/4	7/8	7/8	*T to S*	*T to S*	*T to S*	1/64
1 3/4	5/8	3/4	3/4	3/4	7/8	1	...	...	...	1/64
2	5/8	3/4	3/4	3/4	7/8	1	1 1/4	1 1/4	1 1/4	1/64
2 1/4	5/8	3/4	3/4	3/4	7/8	1	...	...	...	1/64
2 1/2	5/8	3/4	3/4	7/8	7/8	1	1 1/4	1 1/4	1 1/4	1/64
2 3/4	3/4	3/4	7/8	7/8	7/8	1	...	...	...	1/64
3	3/4	3/4	7/8	7/8	7/8	1 1/8	1 3/8	1 3/8	1 1/2	1/64
3 1/4	3/4	3/4	7/8	7/8	1	1 1/8	...	...	...	1/32
3 1/2	3/4	3/4	7/8	7/8	1	1 1/8	1 3/8	1 3/8	1 1/2	1/32
3 3/4	..	3/4	7/8	7/8	1	1 1/8	...	...	...	1/32
4	..	7/8	7/8	7/8	1	1 1/8	1 1/2	1 1/2	1 1/2	1/32
4 1/2	..	7/8	1	1	1 1/8	1 1/4	1 1/2	1 1/2	1 5/8	1/32
5	..	..	1	1	1 1/8	1 1/4	1 1/2	1 1/2	1 5/8	1/32
5 1/2	..	..	1 1/8	1 1/8	1 1/8	1 1/4	1 5/8	1 5/8	1 3/4	1/32
6	..	..	1 1/8	1 1/8	1 1/8	1 1/4	1 5/8	1 5/8	1 3/4	1/32
6 1/2	..	..	1 1/8	1 1/8	1 1/8	1 1/4	1 5/8	1 5/8	1 3/4	1/16
7	..	..	...	1 1/8	1 1/4	1 1/4	1 5/8	1 5/8	1 3/4	1/16
7 1/2	..	..	...	1 1/8	1 1/4	1 1/4	1 5/8	1 5/8	1 3/4	1/16
8	..	..	...	1 1/4	1 1/4	1 3/8	1 5/8	1 5/8	1 7/8	1/16
8 1/2	..	..	...	...	1 1/4	1 3/8	1 3/4	1 3/4	1 7/8	1/16
9	..	..	...	...	1 1/4	1 3/8	1 3/4	1 3/4	2	1/16
9 1/2	..	..	...	...	1 1/4	1 3/8	1 3/4	1 3/4	2	1/16
10	..	..	...	...	1 1/4	1 3/8	1 3/4	1 3/4	2	1/16

T to S means threaded to square as near as is practicable
Radius of fillet between body and head 1/32 in. on sizes No. 10 to 1/2 in., inclusive, and 1/16 in. on sizes 9/16, 5/8 and 3/4 in.

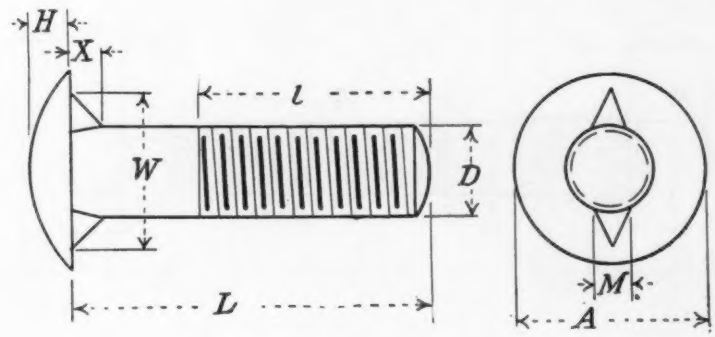


TABLE 2—FIN-NECK CARRIAGE-BOLT

Nominal Size	D: Major Diameter of Thread, Maximum Basic	D: Major Diameter of Thread, Tolerance −	Threads per Inch	A: Diameter of Head, Basic	A: Diameter of Head, Tolerance + or −	H: Height of Head, Basic	H: Height of Head, Tolerance + or −	X: Depth of Fins, Basic	X: Depth of Fins, Tolerance +	W: Distance across Fins, Basic	W: Distance across Fins, Tolerance + or −	M: Maximum Thickness of Fins, Basic	M: Maximum Thickness of Fins, Tolerance + or −
No. 10	0.190	0.009	24	0.469 (15/32)	0.010	0.078	0.010	0.078	0.010	0.375	0.010	0.078	0.010
1/4	0.250	0.010	20	0.594 (19/32)	0.010	0.109	0.010	0.094	0.010	0.438	0.010	0.094	0.010
5/16	0.313	0.013	18	0.719 (23/32)	0.010	0.141	0.010	0.125	0.010	0.531	0.010	0.125	0.010
3/8	0.375	0.015	16	0.844 (27/32)	0.010	0.172	0.010	0.141	0.010	0.625	0.010	0.141	0.010
7/16	0.438	0.015	14	0.969 (31/32)	0.010	0.203	0.010	0.172	0.010	0.719	0.010	0.172	0.010
1/2	0.500	0.015	13	1.094 (1 3/32)	0.010	0.234	0.010	0.188	0.010	0.813	0.010	0.188	0.010

Nominal Size	No. 10	1/4	5/16	3/8	7/16	1/2	Length Tolerance + or −
Length of Bolt, L	Minimum Length of Thread, l, Including Point						
1/2	*T to F*	*T to F*	...	...	...	...	1/64
3/4	3/8	3/8	3/8	3/8	...	...	1/64
1	1/2	1/2	1/2	5/8	5/8	*T to F*	1/64
1 1/4	1/2	5/8	5/8	5/8	3/4	3/4	1/64
1 1/2	1/2	3/4	3/4	3/4	7/8	7/8	1/64
1 3/4	5/8	3/4	3/4	3/4	7/8	1	1/64
2	5/8	3/4	3/4	3/4	7/8	1	1/64
2 1/4	5/8	3/4	3/4	3/4	7/8	1	1/64
2 1/2	5/8	3/4	3/4	3/4	7/8	1	1/64
2 3/4	3/4	3/4	7/8	7/8	7/8	1	1/64
3	3/4	3/4	7/8	7/8	7/8	1 1/8	1/64
3 1/4	3/4	3/4	7/8	7/8	1	1 1/8	1/32
3 1/2	3/4	3/4	7/8	7/8	1	1 1/8	1/32
3 3/4	....	3/4	7/8	7/8	1	1 1/8	1/32
4	....	7/8	7/8	7/8	1	1 1/8	1/32
4 1/2	....	7/8	7/8	1	1 1/8	1 1/4	1/32
5	....	....	1	1	1 1/8	1 1/4	1/32
5 1/2	....	....	1 1/8	1 1/8	1 1/8	1 1/4	1/32
6	....	....	1 1/8	1 1/8	1 1/8	1 1/4	1/32
6 1/2	....	....	1 1/8	1 1/8	1 1/8	1 1/4	1/16
7	....	....	.....	1 1/8	1 1/4	1 1/4	1/16
7 1/2	....	....	.....	1 1/8	1 1/4	1 1/4	1/16
8	....	....	.....	1 1/4	1 1/4	1 3/8	1/16
8 1/2	....	....	.....	.....	1 1/4	1 3/8	1/16

T to F means threaded to fin as near as is practicable
Radius of fillet between body and head 1/32 in.

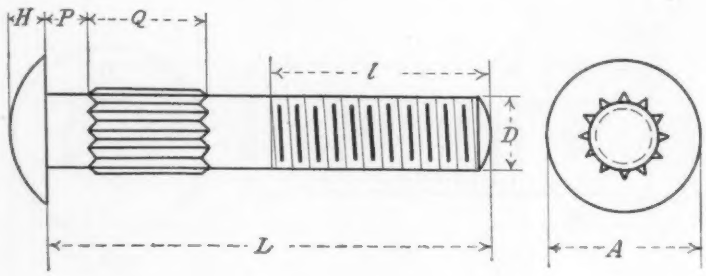


TABLE 3—RIBBED CARRIAGE-BOLT

Nominal Size	D		Threads per Inch	A		H		P		Q		Maximum Number of Ribs	Included Angle of Rib
	Major Diameter of Thread			Diameter of Head		Height of Head		Distance of Rib Below Head		Length of Rib			
	Maximum Basic	Tolerance −		Basic	Tolerance + or −	Basic	Tolerance + or −	Basic	Tolerance + or −	When L Is 1 1/8 In. and Under	When L Is 1 1/4 In. and Over		
No. 10	0.190	0.009	24	0.438 (7/16)	0.010	0.094	0.010	0.094	0.031	0.375	0.500	9	Approximately 90 Deg.
1/4	0.250	0.010	20	0.563 (9/16)	0.010	0.125	0.010	0.094	0.031	0.375	0.500	10	
5/16	0.313	0.013	18	0.688 (11/16)	0.010	0.156	0.010	0.094	0.031	0.375	0.500	12	
3/8	0.375	0.015	16	0.813 (13/16)	0.010	0.188	0.010	0.094	0.031	0.375	0.500	12	
7/16	0.438	0.015	14	0.938 (15/16)	0.010	0.219	0.010	0.094	0.031	0.375	0.500	14	
1/2	0.500	0.015	13	1.063 (1 1/16)	0.010	0.250	0.010	0.094	0.031	0.375	0.500	16	

Nominal Size	No. 10	1/4	5/16	3/8	7/16	1/2	Length Tolerance + or −
Length of Bolt, L	Minimum Length of Thread, l, Including Point						
1	*T to R*	*T to R*	*T to R*	*T to R*	*T to R*	*T to R*	1/64
1 1/4	1/2	5/8	5/8	*T to R*	*T to R*	*T to R*	1/64
1 1/2	1/2	3/4	3/4	3/4	*T to R*	*T to R*	1/64
1 3/4	5/8	3/4	3/4	3/4	7/8	1	1/64
2	5/8	3/4	3/4	3/4	7/8	1	1/64
2 1/4	5/8	3/4	3/4	3/4	7/8	1	1/64
2 1/2	5/8	3/4	3/4	3/4	7/8	1	1/64
2 3/4	3/4	3/4	7/8	7/8	7/8	1	1/64
3	3/4	3/4	7/8	7/8	7/8	1 1/8	1/64
3 1/4	3/4	3/4	7/8	7/8	1	1 1/8	1/32
3 1/2	3/4	3/4	7/8	7/8	1	1 1/8	1/32
3 3/4	....	3/4	7/8	7/8	1	1 1/8	1/32
4	....	7/8	7/8	7/8	1	1 1/8	1/32
4 1/2	....	7/8	1	1	1 1/8	1 1/4	1/32
5	....	....	1	1	1 1/8	1 1/4	1/32
5 1/2	....	....	1 1/8	1 1/8	1 1/8	1 1/4	1/32
6	....	....	1 1/8	1 1/8	1 1/8	1 1/4	1/32
6 1/2	....	....	1 1/8	1 1/8	1 1/8	1 1/4	1/16
7	....	....	.....	1 1/8	1 1/4	1 1/4	1/16
7 1/2	....	....	.....	1 1/8	1 1/4	1 1/4	1/16
8	....	....	.....	1 1/4	1 1/4	1 3/8	1/16
8 1/2	....	....	.....	.....	1 1/4	1 3/8	1/16
9	....	....	.....	.....	1 1/4	1 3/8	1/16
9 1/2	....	....	.....	.....	1 1/4	1 3/8	1/16
10	....	....	.....	.....	1 1/4	1 3/8	1/16

T to R means threaded to rib as near as is practicable
Radius of fillet between body and head 1/32 in.

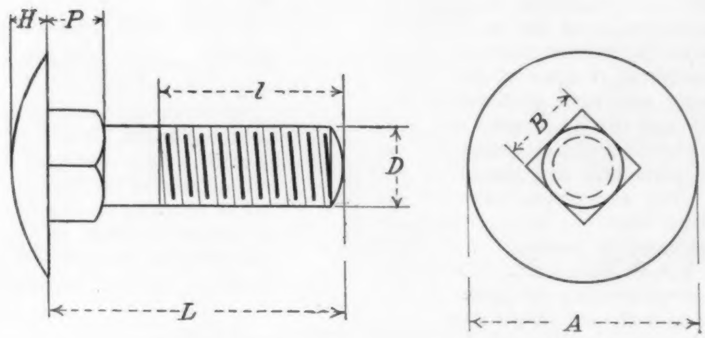


TABLE 4—STEP-BOLT

Nominal Size	D: Major Diameter of Thread — Maximum Basic	D: Tolerance −	Threads per Inch	A: Diameter of Head — Basic	A: Tolerance + or −	H: Height of Head — Basic	H: Tolerance + or −	P: Depth of Square — Basic	P: Tolerance +	B: Width of Square — Basic	B: Tolerance −
No. 10	0.190	0.009	24	0.625 (5/8)	0.010	0.094	0.010	0.188	0.031	0.190	0.009
1/4	0.250	0.010	20	0.813 (13/16)	0.010	0.125	0.010	0.219	0.031	0.250	0.010
5/16	0.313	0.013	18	1.000 (1)	0.010	0.156	0.010	0.250	0.031	0.313	0.013
3/8	0.375	0.015	16	1.188 (1 3/16)	0.010	0.188	0.010	0.281	0.031	0.375	0.015
7/16	0.438	0.015	14	1.375 (1 3/8)	0.010	0.219	0.010	0.313	0.031	0.438	0.015
1/2	0.500	0.015	13	1.563 (1 9/16)	0.010	0.250	0.010	0.344	0.031	0.500	0.015

Nominal Size / Length of Bolt, L	No. 10	1/4	5/16	3/8	7/16	1/2	Length Tolerance + or −
	Minimum Length of Thread, l, Including Point						
1/2	*T to S*	*T to S*	*T to S*	.....	.....	.....	1/64
3/4	*T to S*	*T to S*	*T to S*	*T to S*	*T to S*	.....	1/64
1	1/2	1/2	1/2	5/8	5/8	*T to S*	1/64
1 1/4	1/2	5/8	5/8	5/8	3/4	3/4	1/64
1 1/2	1/2	3/4	3/4	3/4	7/8	7/8	1/64
1 3/4	5/8	3/4	3/4	3/4	7/8	1	1/64
2	5/8	3/4	3/4	3/4	7/8	1	1/64
2 1/4	5/8	3/4	3/4	3/4	7/8	1	1/64
2 1/2	5/8	3/4	3/4	3/4	7/8	1	1/64
2 3/4	3/4	3/4	7/8	7/8	7/8	1	1/64
3	3/4	3/4	7/8	7/8	7/8	1 1/8	1/64
3 1/4	3/4	3/4	7/8	7/8	1	1 1/8	1/32
3 1/2	3/4	3/4	7/8	7/8	1	1 1/8	1/32
3 3/4	....	3/4	7/8	7/8	1	1 1/8	1/32
4	....	7/8	7/8	7/8	1	1 1/8	1/32
4 1/2	....	7/8	1	1	1 1/8	1 1/4	1/32
5	....	....	1	1	1 1/8	1 1/4	1/32
5 1/2	....	....	1 1/8	1 1/8	1 1/8	1 1/4	1/32
6	....	....	1 1/8	1 1/8	1 1/8	1 1/4	1/32
6 1/2	....	....	1 1/8	1 1/8	1 1/8	1 1/4	1/16
7	....	....	.....	1 1/8	1 1/4	1 1/4	1/16
7 1/2	....	....	.....	1 1/8	1 1/4	1 1/4	1/16
8	....	....	.....	1 1/4	1 1/4	1 3/8	1/16
8 1/2	....	....	.....	.....	1 1/4	1 3/8	1/16

T to S means threaded to square as near as is practicable
Radius of fillet between body and head 1/32 in.

In accordance with the Society's regular procedure the report was assigned to the Screw-Threads Division of the Standards Committee and submitted to the members of the Division some time before its meeting in Detroit on Dec. 16. The report was approved for adoption as tentative American Standard under the procedure of the American Engineering Standards Committee, with the exception of the second table for each type of bolt that refers to the bolt lengths and thread lengths. It was felt that the principal feature of the report is the standardization of the head and neck dimensions and body diameters but that the body and thread lengths are governed more by the requirements of individual design of apparatus in which the bolts are used and are therefore not susceptible of standardization. The recommendation of the Division in this respect is in line with the action that has been taken by it in similar instances in passing on reports for adoption by the Society as S.A.E. Standard. The Division also felt that if further consideration is given to including these tables of the body and thread lengths on bolts, such dimensions should be correlated with those for all other similar screw thread products, taking into consideration existing standards. The report is printed in full in this issue of THE JOURNAL for review by the automotive industry prior to action being taken on it by the Standards Committee at the Annual Meeting of the Society in Detroit, Jan. 25 to 28.

ROUND UNSLOTTED-HEAD BOLTS

The following notes refer to all of the various types of this class of bolt:

All screw threads are to be American Standard, Coarse-Thread Series, Free Fit (Class 2), with special major-diameter tolerances provided for unfinished hot-rolled material.

The threads on these bolts shall be produced by cutting or rolling. When rolled, the shank diameter will necessarily be smaller than the shank diameter for corresponding cut threads.

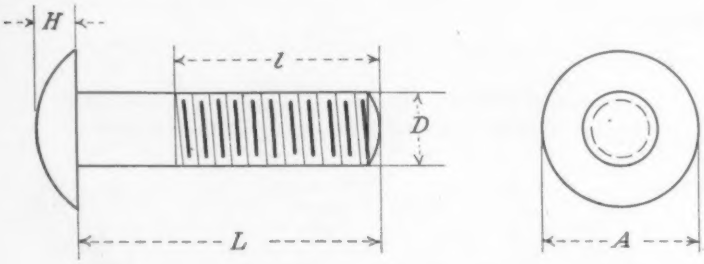


TABLE 5—BUTTON-HEAD MACHINE-BOLT

Nominal Size	D		Threads per Inch	A		H	
	Major Diameter of Thread			Diameter of Head		Height of Head	
	Maximum Basic	Tolerance —		Basic	Tolerance + or −	Basic	Tolerance + or −
No. 10	0.190	0.009	24	0.438 (7/16)	0.010	0.094	0.010
¼	0.250	0.010	20	0.563 (9/16)	0.010	0.125	0.010
5/16	0.313	0.013	18	0.688 (11/16)	0.010	0.156	0.010
⅜	0.375	0.015	16	0.813 (13/16)	0.010	0.188	0.010
7/16	0.438	0.015	14	0.938 (15/16)	0.010	0.219	0.010
½	0.500	0.015	13	1.063 (1 1/16)	0.010	0.250	0.010
9/16	0.563	0.016	12	1.188 (1 3/16)	0.015	0.281	0.015
⅝	0.625	0.017	11	1.313 (1 5/16)	0.015	0.313	0.015
¾	0.750	0.020	10	1.563 (1 9/16)	0.015	0.375	0.015

Nominal Size	No. 10	¼	5/16	⅜	7/16	½	9/16	⅝	¾	Length Tolerance + or −
Length of Bolt, L	Minimum Length of Thread, l, Including Point.									
½	⅜	⅜	⅜	...	...	...	...	...	...	1/64
¾	⅜	⅜	⅜	⅜	½	...	...	...	...	1/64
1	½	½	½	⅝	⅝	¾	...	...	...	1/64
1¼	½	⅝	⅝	⅝	¾	¾	1	1	...	1/64
1½	½	¾	¾	¾	⅞	⅞	1⅛	1⅛	1⅛	1/64
1¾	⅝	¾	¾	¾	⅞	1	...	...	...	1/64
2	⅝	¾	¾	¾	⅞	1	1¼	1¼	1¼	1/64
2¼	⅝	¾	¾	¾	⅞	1	...	...	...	1/64
2½	⅝	¾	¾	⅞	⅞	1	1¼	1¼	1¼	1/64
2¾	¾	¾	⅞	⅞	⅞	1	...	...	...	1/64
3	¾	¾	⅞	⅞	⅞	1⅛	1⅜	1⅜	1½	1/64
3¼	¾	¾	⅞	⅞	1	1⅛	...	...	...	1/32
3½	¾	¾	⅞	⅞	1	1⅛	1⅜	1⅜	1½	1/32
3¾	..	¾	⅞	⅞	1	1⅛	...	...	...	1/32
4	..	⅞	⅞	⅞	1	1⅛	1½	1½	1½	1/32
4½	..	⅞	1	1	1⅛	1¼	1½	1½	1⅝	1/32
5	..	..	1	1	1⅛	1¼	1½	1½	1⅝	1/32
5½	..	..	1⅛	1⅛	1⅛	1¼	1⅝	1⅝	1¾	1/32
6	..	..	1⅛	1⅛	1⅛	1¼	1⅝	1⅝	1¾	1/32
6½	..	..	1⅛	1⅛	1⅛	1¼	1⅝	1⅝	1¾	1/16
7	..	..	...	1⅛	1¼	1¼	1⅝	1⅝	1¾	1/16
7½	..	..	...	1⅛	1¼	1¼	1⅝	1⅝	1¾	1/16
8	..	..	...	1¼	1¼	1⅜	1⅝	1⅝	1⅞	1/16
8½	..	..	...	...	1¼	1⅜	1¾	1¾	1⅞	1/16
9	..	..	...	...	1¼	1⅜	1¾	1¾	2	1/16
9½	..	..	...	...	1¼	1⅜	1¾	1¾	2	1/16
10	..	..	...	...	1¼	1⅜	1¾	1¾	2	1/16

Radius of fillet between body and head 1/32 in. on sizes No. 10 to ½ in., inclusive, and 1/16 in. on sizes 9/16, ⅝ and ¾ in.

SET-SCREW HEADS AND JAM NUTS

Screw Threads Division Submits Tables for Approval as S.A.E. Standard

After the Society had approved the present S.A.E. Standard for Wrench-Head Bolts and Nuts and Wrench Openings last June, Table 4, for Set-Screw Heads, p. C6c, and Table 7, for Finished and Semi-Finished Jam Nuts, p. C6f, were included in the S.A.E. HANDBOOK as General Information only, as they had not been regularly approved.

The complete report, as acted on by the Society at that time, and Tables 4 and 7 were taken from the report of the Sectional Committee on the Standardization of Bolt, Nut and Rivet Proportions for which this Society and the American Society of Mechanical Engineers are sponsors under the procedure of the American Engineering Standards Committee. The Screw Threads Division has since then approved these tables and recommends their adoption by the Society for inclusion in the S.A.E. Standard for Wrench-Head Bolts and Nuts and Wrench Openings commencing on p. C6 of the S.A.E. HANDBOOK.

TABLE 4—SET-SCREW HEADS

Diameter of Screw D		Width Across Flats			Height		
		Maximum		Minimum	Nominal	Maximum	Minimum
¼	0.2500	¼	0.2500	0.241	3/16	0.197	0.179
5/16	0.3125	5/16	0.3125	0.302	15/64	0.245	0.224
⅜	0.3750	⅜	0.3750	0.362	9/32	0.293	0.270
7/16	0.4375	7/16	0.4375	0.423	21/64	0.341	0.315
½	0.5000	½	0.5000	0.484	⅜	0.389	0.361
9/16	0.5625	9/16	0.5625	0.545	27/64	0.437	0.407
⅝	0.6250	⅝	0.6250	0.607	15/32	0.485	0.452
¾	0.7500	¾	0.7500	0.729	9/16	0.582	0.544
⅞	0.8750	⅞	0.8750	0.852	21/32	0.678	0.635
1	1.0000	1	1.0000	0.974	¾	0.774	0.726
1⅛	1.1250	1⅛	1.1250	1.097	27/32	0.870	0.817
1¼	1.2500	1¼	1.2500	1.219	15/16	0.967	0.909
1½	1.5000	1½	1.5000	1.464	1⅛	1.159	1.091

FORMULAS

Width across flats of set-screw heads shall be equal to diameter of screw, D.

Minimum width across rounded corners of square equals 1.373 times minimum width across flats.

Tolerance for width across flats shall be minus $0.020\,D + 0.006$ (except for 1/4 and 5/16).

Height of set-screw heads shall be $\frac{3}{4}\,D$.

Tolerance for height of heads shall be $0.040\,D + 0.008$ from the minimum.

Width of neck under head shall not be over twice the pitch of the thread.

The radius of the crown of the head shall be $2\frac{1}{2}\,D$.

When head is not necked it shall be beveled not more than 40 deg. under the head.

(*Concluded on p. 34*)

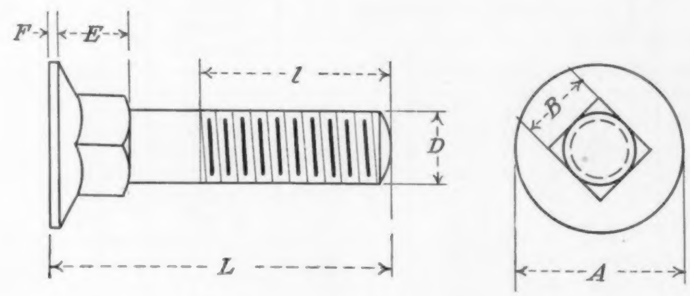


TABLE 6—COUNTERSUNK CARRIAGE-BOLT

Nominal Size	*D*		Threads per Inch	*A*		*F*	Included Angle, Deg.	*E*		*B*	
	Major Diameter of Thread			Diameter of Head		Feed Thickness		Depth of Square		Width of Square	
	Maximum Basic	Tolerance −		Basic	Tolerance + or −			Basic	Tolerance +	Basic	Tolerance −
No. 10	.....	.....	.....	....	.....	.....	.....	.....	.....	.....	.....
¼	0.250	0.010	20	0.625 (⅝)	0.010	0.016	114	0.281	0.031	0.250	0.010
5/16	0.313	0.013	18	0.750 (¾)	0.010	0.031	114	0.344	0.031	0.313	0.013
⅜	0.375	0.015	16	0.875 (⅞)	0.010	0.031	114	0.406	0.031	0.375	0.015
7/16	0.438	0.015	14	1.000 (1)	0.010	0.031	114	0.469	0.031	0.438	0.015
½	0.500	0.015	13	1.125 (1⅛)	0.010	0.031	114	0.531	0.031	0.500	0.015
9/16	0.563	0.016	12	1.250 (1¼)	0.015	0.031	114	0.594	0.031	0.563	0.016
⅝	0.625	0.017	11	1.375 (1⅜)	0.015	0.031	114	0.656	0.031	0.625	0.017
¾	0.750	0.020	10	1.625 (1⅝)	0.015	0.047	114	0.781	0.031	0.750	0.020

Nominal Size	No. 10	¼	5/16	⅜	7/16	½	9/16	⅝	¾	Length Tolerance + or −
Length of Bolt, *L*	Minimum Length of Thread, *l*, Including Point									
¾	*T to S*	*T to S*	*T to S*	*T to S*	*T to S*	.....	.....	.....	.....	1/64
1	½	½	½	*T to S*	*T to S*	*T to S*	.....	.....	.....	1/64
1¼	½	⅝	⅝	⅝	*T to S*	*T to S*	*T to S*	*T to S*	.....	1/64
1½	½	¾	¾	¾	⅞	*T to S*	*T to S*	*T to S*	*T to S*	1/64
1¾	⅝	¾	¾	¾	⅞	1	.....	.....	.....	1/64
2	⅝	¾	¾	¾	⅞	1	1¼	*T to S*	*T to S*	1/64
2¼	⅝	¾	¾	⅞	⅞	1	.....	.....	.....	1/64
2½	⅝	¾	¾	⅞	⅞	1	1¼	1¼	1¼	1/64
2¾	¾	¾	⅞	⅞	⅞	1	.....	.....	.....	1/64
3	¾	¾	⅞	⅞	⅞	1⅛	1⅜	1⅜	1½	1/64
3¼	¾	¾	⅞	⅞	1	1⅛	.....	.....	.....	1/32
3½	¾	¾	⅞	⅞	1	1⅛	1⅜	1⅜	1½	1/32
3¾	.....	¾	⅞	⅞	1	1⅛	.....	.....	.....	1/32
4	....	⅞	⅞	⅞	1	1⅛	1½	1½	1½	1/32
4½	....	⅞	1	1	1⅛	1¼	1½	1½	1⅝	1/32
5	....	....	1	1	1⅛	1¼	1½	1½	1⅝	1/32
5½	....	....	1⅛	1⅛	1⅛	1¼	1⅝	1⅝	1¾	1/32
6	....	....	1⅛	1⅛	1⅛	1¼	1⅝	1⅝	1¾	1/32
6½	....	....	1⅛	1⅛	1⅛	1¼	1⅝	1⅝	1¾	1/16
7	....	....	.....	1⅛	1¼	1¼	1⅝	1⅝	1¾	1/16
7½	....	....	.....	1⅛	1¼	1¼	1⅝	1⅝	1¾	1/16
8	....	....	.....	1¼	1¼	1⅜	1⅝	1⅝	1⅞	1/16
8½	....	....	.....	.....	1¼	1⅜	1¾	1⅝	1⅞	1/16
9	....	....	.....	.....	1¼	1⅜	1¾	1¾	2	1/16
9½	....	....	.....	.....	1¼	1⅜	1¾	1¾	2	1/16
10	....	....	.....	.....	1¼	1⅜	1¾	1¾	2	1/16

T to S means threaded to square as near as is practicable
Radius of fillet between body and head 1/32 in. on sizes No.10 to ½ in., inclusive, and 1/16 in. on sizes 9/16, ⅝ and ¾ in.

SCREW HEADS AND JAM NUTS (*Concluded from p. 32*)

TABLE 7—FINISHED AND SEMI-FINISHED JAM-NUTS

Diameter of Bolt		Width across Flats			Minimum Width Across Corners of Hexagon	Thickness		
Nominal	Maximum	Nominal	Maximum	Minimum		Nominal	Maximum	Minimum
1/4	0.2500	7/16	0.4375	0.428	0.488	5/32	0.163	0.150
5/16	0.3125	9/16	0.5625	0.552	0.628	3/16	0.195	0.180
3/8	0.3750	5/8	0.6250	0.613	0.699	7/32	0.227	0.211
7/16	0.4375	3/4	0.7500	0.737	0.840	1/4	0.259	0.241
1/2	0.5000	13/16	0.8125	0.799	0.911	5/16	0.323	0.303
9/16	0.5625	7/8	0.8750	0.860	0.980	11/32	0.355	0.333
5/8	0.6250	15/16	0.9375	0.922	1.051	3/8	0.387	0.363
3/4	0.7500	1 1/8	1.1250	1.108	1.263	7/16	0.451	0.424
7/8	0.8750	1 5/16	1.3125	1.293	1.474	1/2	0.516	0.484
1	1.0000	1 1/2	1.5000	1.479	1.686	9/16	0.580	0.545
1 1/8	1.1250	1 11/16	1.6875	1.665	1.898	5/8	0.644	0.606
1 1/4	1.2500	1 7/8	1.8750	1.850	2.109	3/4	0.771	0.729
1 1/2	1.5000	2 1/4	2.2500	2.222	2.538	7/8	0.900	0.850
1 3/4	1.7500	2 5/8	2.6250	2.593	2.956	1	1.029	0.971
2	2.0000	3	3.0000	2.964	3.379	1 1/8	1.158	1.093
2 1/4	2.2500	3 3/8	3.3750	3.335	3.802	1 1/4	1.286	1.214
2 1/2	2.5000	3 3/4	3.7500	3.707	4.226	1 1/2	1.540	1.460
2 3/4	2.7500	4 1/8	4.1250	4.078	4.649	1 5/8	1.669	1.581
3	3.0000	4 1/2	4.5000	4.449	5.072	1 3/4	1.798	1.703

The finished top or both top and bottom of jam-nuts shall be flat and chamfered; angle of chamfer with surface 30 deg.; diameter of top, or both top and bottom circles shall be 100 per cent of the nominal width across flats. Tolerance on top flat surface shall be minus 15 per cent. For jam-nuts with a washer-face, the thickness of the washer-face shall be 1/64 in. The bearing-surface of washer-face shall be 100 per cent of the nominal width across flats. Tolerance on the diameter of the washer-face shall be plus or minus 5 per cent. The thickness of the nut shall be measured from the top of the nut to the bearing-surface.

The axis of the threaded hole shall be at right angles to the washer-face within a tolerance of 2 deg.

Width across flats of rough and semi-finished regular nuts shall be 1½ D except as follows:

Diameter of bolt, 1/4 to 9/16; width across flats, 1½ $D + 1/16$ with adjustment in the 16th-in. sizes to eliminate 32nd-in. size wrench openings.

Tolerance for width across flats shall be minus 0.05 D.

Minimum width across rounded corners of hexagon equals 1.14 times minimum width across flats.

Minimum width across rounded corners of square equals 1.373 times minimum width across flats.

Thickness for rough and semi-finished regular nuts shall be 7/8 D.

Tolerance for thickness shall be 0.032 $D + 0.024$ from the minimum.

The top of rough and semi-finished regular square and hexagonal nuts shall be flat and chamfered; angle of chamfer with surface shall be 30 deg.; diameter of top, or of both top and bottom circle, shall be 100 per cent of the nominal width across flats.

Tolerance on diameter of top flat circle shall be minus 15 per cent.

Semi-finished nuts shall be faced on bearing-surface and at right angles to the axis of the thread within 3 deg.

Width across flats to be measured at the top of the nut.

Taper of sides of nuts shall not exceed 4 deg.

T-SLOTS, BOLTS, NUTS AND CUTTERS

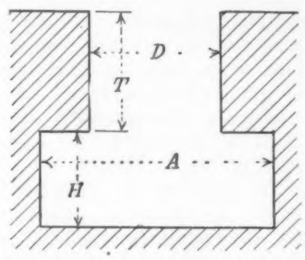
Production Division Approves Sectional Committee Report on These Subjects

This report, which is the first one issued by the Sectional Committee on Small Tools and Machine-Tool Elements, was formulated by the following Subcommittee:

Erik Oberg, *Chairman*	*Machinery*
Herman Casler, *Secretary*	Canastota, N. Y.
Joseph B. Armitage	Kearney & Trecker Corporation
Joseph W. C. Bullard	Bullard Machine Tool Co.
Luther D. Burlingame	Brown & Sharpe Mfg. Co.
Edward P. Burrell	Warner & Swasey
Harry Cadwallader, Jr.	Standard Shop Equipment Co.
Robert T. Hazelton	Cincinnati
E. G. Herndon	Navy Department
LeRoy F. Maurer	Studebaker Corporation of America
Frank O. Hoagland	Pratt & Whitney Co.
George Langen	Cincinnati Planer Co.
Edson R. Norris	Westinghouse Electric & Mfg. Co.
Charles W. Thomas	Columbia University

This Subcommittee was organized in 1924 and held a series of meetings, at which data collected as the basis for the standard were drafted into a preliminary report. This was submitted to the Production Division of the S.A.E. Standards Committee for criticism and subsequently revised and approved by the Sectional Committee and submitted to the American Society of Mechanical Engineers, the National Machine Tool Builders' Association and this Society for approval as sponsors.

When the report has been approved and adopted it will be of much value to machine-tool builders and users in unifying the individual and varying standards that have come into use by machine-tool builders, some of which have been based on the strength of ordinary gray iron used in the table castings or other places where T-slots are located, and of comparatively low-grade steel for the bolts or studs. Other T-slots have been based on harder table material and on bolts and studs made from stronger stock. In preparing the report, consideration has been given to the factors that should control standardization, such as providing sufficient clearance for oil and chips and for free sliding of the T-bolts in their slots. The Sectional Committee feels that the



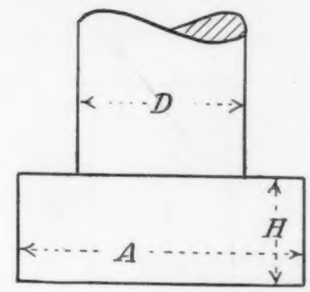


TABLE 1—PROPOSED DIMENSIONS FOR T-SLOTS

Diameter of T-Bolt[1]	Width of Throat[1,2] D	Depth of Throat T: Maximum	Depth of Throat T: Minimum	Head Space Dimensions and Tolerances — Width A: Maximum Basic	Width A: Tolerance Minus	Width A: Minimum	Depth H: Maximum Basic	Depth H: Tolerance Minus	Depth H: Minimum
1/4	9/32	3/8	1/8	9/16	0.063	1/2	15/64	0.031	13/64
5/16	11/32	7/16	5/32	21/32	0.063	19/32	17/64	0.031	15/64
3/8	7/16	9/16	7/32	25/32	0.063	23/32	21/64	0.031	19/64
1/2	9/16	11/16	5/16	31/32	0.063	29/32	25/64	0.031	23/64
5/8	11/16	7/8	7/16	1 1/4	0.063	1 3/16	31/64	0.031	29/64
3/4	13/16	1 1/16	9/16	1 15/32	0.094	1 3/8	5/8	0.031	19/32
1	1 1/16	1 1/4	3/4	1 27/32	0.094	1 3/4	53/64	0.047	25/32
1 1/4	1 5/16	1 9/16	1	2 7/32	0.094	2 1/8	1 3/32	0.063	1 1/32
1 1/2	1 9/16	1 15/16	1 1/4	2 21/32	0.094	2 9/16	1 11/32	0.063	1 9/32

[1] In addition to the "width of throat" given above, a secondary standard is recognized, having the "width of throat" the same as the nominal diameter of the T-bolt. This is to provide for the use during the transition period of this standard on many machine-tools where it is already established.

"Width of tongue" to be used with the above T-slots will be found in Table 5.

[2] A tolerance of minus 0.001 is allowed for "width of throat" when tongues or other parts must fit.

TABLE 2—PROPOSED DIMENSIONS FOR T-BOLTS

Diameter of T-Bolt[1,2] D	Threads per In.[1]	Bolt Head Dimensions and Tolerances — Width Across Flats A: Maximum Basic	Width Across Flats A: Tolerance Minus	Width Across Flats A: Minimum	Width Across Corners	Height H: Maximum Basic	Height H: Tolerance Minus	Height H: Minimum
1/4	20	15/32	0.031	7/16	0.663	5/32	0.016	9/64
5/16	18	9/16	0.031	17/32	0.796	3/16	0.016	11/64
3/8	16	11/16	0.031	21/32	0.972	1/4	0.016	15/64
1/2	13	7/8	0.031	27/32	1.238	5/16	0.016	19/64
5/8	11	1 1/8	0.031	1 3/32	1.591	13/32	0.016	25/64
3/4	10	1 5/16	0.031	1 9/32	1.856	17/32	0.031	1/2
1	8	1 11/16	0.031	1 21/32	2.387	11/16	0.031	21/32
1 1/4	7	2 1/16	0.031	2 1/32	2.917	15/16	0.031	29/32
1 1/2	6	2 1/2	0.031	2 15/32	3.536	1 3/16	0.031	1 5/32

[1] Tolerances for diameters of bolts or studs and for threads are in accordance with the American Standard Screw Threads, Coarse Thread series, Medium Fit (Class 3). If a free or close fit thread is desired the tolerances given in the American Standard Screw Threads for either of these classes of fit shall be followed.

[2] T-slots to be used with these bolts will be found in Table 1.

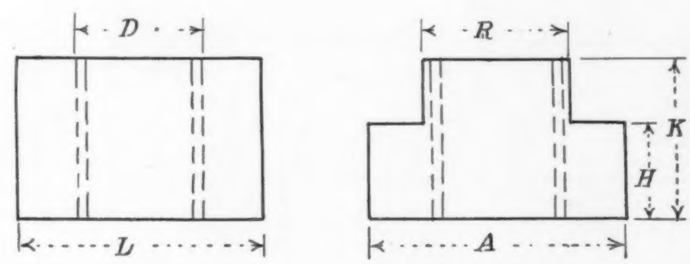


TABLE 3—PROPOSED DIMENSIONS FOR T-NUTS

Tap for Stud[1] D: Diameter	Tap for Stud D: Threads per In.	Width of Throat T-Slot	Width of Tongue R: Maximum Basic	Width of Tongue R: Tolerance Minus	Width of Tongue R: Minimum	Width of Nut[2] A: Maximum Basic	Width of Nut A: Tolerance Minus	Width of Nut A: Minimum	Height of Nut[2] H: Maximum Basic	Height of Nut H: Tolerance Minus	Height of Nut H: Minimum	Total Thickness, Including Tongue K	Length of Nut[3] L
1/4	20	11/32	0.330	0.010	0.320	9/16	0.031	17/32	3/16	0.016	11/64	9/32	9/16
5/16	18	7/16	0.418	0.010	0.408	11/16	0.031	21/32	1/4	0.016	15/64	3/8	11/16
3/8	16	9/16	0.543	0.010	0.533	7/8	0.031	27/32	5/16	0.016	19/64	17/32	7/8
1/2	13	11/16	0.668	0.010	0.658	1 1/8	0.031	1 3/32	13/32	0.016	25/64	5/8	1 1/8
5/8	11	13/16	0.783	0.010	0.773	1 5/16	0.031	1 9/32	17/32	0.031	1/2	25/32	1 5/16
3/4	10	1 1/16	1.033	0.015	1.018	1 11/16	0.031	1 21/32	11/16	0.031	21/32	1	1 11/16
1	8	1 5/16	1.273	0.015	1.258	2 1/16	0.031	2 1/32	15/16	0.031	29/32	1 5/16	2 1/16
1 1/4	7	1 9/16	1.523	0.015	1.508	2 1/2	0.031	2 15/32	1 3/16	0.031	1 5/32	1 5/8	2 1/2

[1] When T-nuts are used, stud *D* is made smaller than the corresponding T-bolt, to assure the full strength of T-nut.
[2] T-slot dimensions to fit the above T-nuts will be found in Table 1.
[3] There are no tolerances given for the "total thickness" or "length of nut" as they need not be held to close limits.

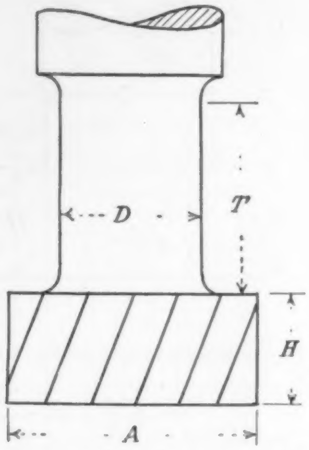


TABLE 4—PROPOSED DIMENSIONS FOR T-SLOT CUTTERS

Width of Throat[1,2]		Thickness of Cutter H		Diameter of Cutters A		Diameter of Neck[2] D	Length of Neck T
Standard	Nominal Bolt Size	Maximum	Minimum Worn	Maximum	Minimum Worn		
9/32	1/4	15/64	13/64	9/16	1/2	17/64	3/8
11/32	5/16	17/64	15/64	21/32	19/32	21/64	7/16
7/16	3/8	21/64	19/64	25/32	23/32	13/32	9/16
9/16	1/2	25/64	23/64	31/32	29/32	17/32	11/16
11/16	5/8	31/64	29/64	1 1/4	1 3/16	21/32	7/8
13/16	3/4	5/8	19/32	1 15/32	1 3/8	25/32	1 1/16
1 1/16	1	53/64	25/32	1 27/32	1 3/4	1 1/32	1 1/4
1 5/16	1 1/4	1 3/32	1 1/32	2 7/32	2 1/8	1 9/32	1 9/16
1 9/16	1 1/2	1 11/32	1 9/32	2 21/32	2 9/16	1 17/32	1 15/16

[1] The "width of throat" given in the above table corresponds to that given in Table 1 on T-slots.
[2] In addition to the "width of throat" given above, a secondary standard is recognized, having the "width of throat" the same as the nominal diameter of the T-bolt. This is to provide for the use, during the transition period, of this standard on many machine tools where it is already established. If the narrower throat is used, the diameter of neck D should be reduced accordingly.

recommendation agrees closely with good practice in machine-tool construction, although the throat of the recommended T-slots is somewhat greater than the nominal diameter of the bolts, as it is believed that this practice should eventually become generally standard. A temporary alternative standard having a throat width equal to the nominal diameter of the bolt is provided, however, for use during the transition period of the standard.

The Production Division of the Standards Committee has approved the report for adoption as tentative American Standard and also for adoption by the Society as S.A.E. Recommended Practice. If so approved by the Society, the report will be the first one relating to production machinery used by the automotive industries that will have been issued by the Production Division.

Production Division Personnel

The members of the Production Division which made this recommendation are:

W. G. Careins, *Chairman*	Ajax Motors Co.
E. N. Sawyer, *Vice-Chairman*	Cleveland Tractor Co.
G. E. Bechtel	White Motor Co.
Eugene Bouton	Chandler-Cleveland Motors Corporation
A. R. Fors	Continental Motors Corporation
A. H. Frauenthal	Chandler-Cleveland Motors Corporation
H. P. Harrison	H. H. Franklin Mfg. Co.
O. C. Kavle	O. C. Kavle Mfg. Engrs.
R. R. Keith	International Harvester Co.
F. E. A. Klein	Pierce-Arrow Motor Car Co.
L. F. Maurer	Studebaker Corporation of America
A. F. Misch	Peerless Motor Car Corporation
W. J. O'Neil	Chrysler Corporation
F. W. Stein	Fairbanks, Morse & Co.
Jas. Theil	Waukesha Motor Co.
E. K. Wennerlund	General Motors Corporation

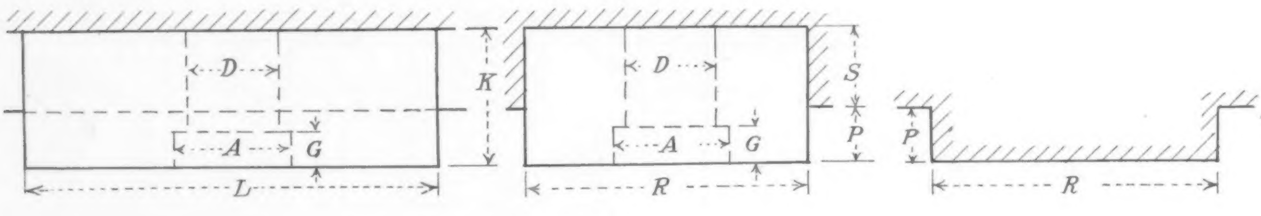


TABLE 5—INSERTED OR SOLID PLAIN TONGUES AND TONGUE SEATS FOR SINGLE WIDTH T-SLOTS

Diameter of T-Bolt[1]	Tongue Dimensions			Depth of Seat	Total Thickness	Screw Dimensions				
	Width[1,2] R	Length[3] L	Projection P	S	K	Diameter of Screw D	Number of Screw	Threads per In.	Diameter of Head A	Thickness of Head G
1/4	9/32	3/8	3/32	1/8	7/32	0.125	5	40	0.196	0.081
5/16	11/32	15/32	1/8	5/32	9/32	0.164	8	32	0.260	0.107
3/8	7/16	9/16	1/8	3/16	5/16	0.190	10	24	0.303	0.124
1/2	9/16	3/4	1/8	7/32	11/32	1/4	.....	20	0.375	0.130
5/8	11/16	15/16	1/8	1/4	3/8	1/4	.....	20	0.375	0.130
3/4	13/16	1 1/8	5/32	9/32	7/16	5/16	.....	18	0.438	0.150
1	1 1/16	1 1/2	7/32	11/32	9/16	3/8	.....	16	0.500	0.170
1 1/4	1 5/16	1 7/8	1/4	3/8	5/8	3/8	.....	16	0.500	0.170
1 1/2	1 9/16	2 1/4	5/16	7/16	3/4	1/2	.....	13	0.625	0.210

[1] In addition to the "width of tongue" given in the above table, a secondary standard is recommended having a width the same as the "nominal diameter of bolt." This is to provide for the use, during the transition period, of this standard on many machine-tools where it is already established.
[2] The "width of tongue" in the above table corresponds to the "width of throat" for T-slots in Table 1.
[3] The "length of tongue" can be varied to suit conditions.

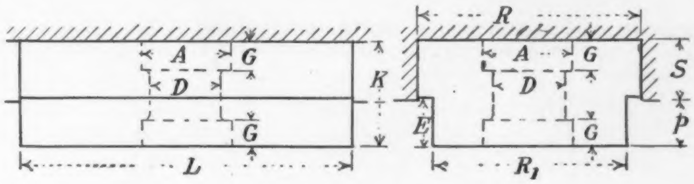


TABLE 6—PROPOSED DIMENSIONS FOR REVERSIBLE TONGUES AND TONGUE SEATS FOR SLOTS FOR TWO SIZES OF T-BOLTS

Diameter of T-Bolt		Tongue Dimensions				Depth of Seat	Total Thickness Including Tongue	Height of Shoulder	Screw Dimensions				
Small	Large	Width[1,2]		Length[3]	Projection				Diameter of Screw	Number of Screw	Threads per In.	Diameter of Head	Thickness of Head
		R_1	R	L	P	S	K	E	D			A	G
1/4	5/16	9/32	11/32	15/32	1/8	5/32	9/32	1/8	0.164	8	32	0.260	0.107
5/16	3/8	11/32	7/16	9/16	1/8	3/16	5/16	9/64	0.190	10	24	0.303	0.124
3/8	1/2	7/16	9/16	3/4	1/8	7/32	11/32	5/32	0.250	.....	20	0.375	0.130
1/2	5/8	9/16	11/16	15/16	1/8	1/4	3/8	5/32	1/4	.....	20	0.375	0.130
5/8	3/4	11/16	13/16	1 1/8	5/32	9/32	7/16	3/16	5/16	.....	18	0.438	0.150
3/4	1	13/16	1 1/16	1 1/2	7/32	11/32	9/16	1/4	3/8	.....	16	0.500	0.170
1	1 1/4	1 1/16	1 5/16	1 7/8	1/4	3/8	5/8	9/32	3/8	.....	16	0.500	0.170
1 1/4	1 1/2	1 5/16	1 9/16	2 1/4	5/16	7/16	3/4	11/32	1/2	.....	13	0.625	0.210

[1] In addition to the "width of tongue" given in the above table, a secondary standard is recommended having a width the same as the "nominal diameter of bolt." This is to provide for the use, during the transition period, of this standard on many machine-tools where it is already established.
The "width of tongue" in the above table corresponds to the the "width of throat" for T-slots in Table 1.
[2] The "length of tongue" can be varied to suit conditions.

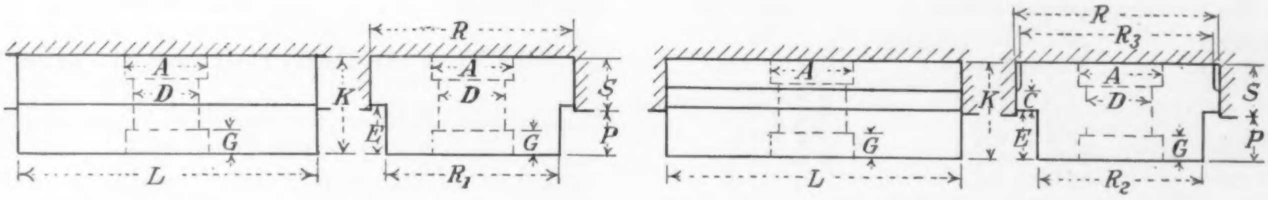


TABLE 7—PROPOSED DIMENSIONS FOR REVERSIBLE TONGUES AND TONGUE SEATS FOR T-SLOTS[1] OF TWO WIDTHS USING THE SAME SIZE T-BOLT

Diameter of T-Bolt	Tongue Dimensions				Depth of Seat	Total Thickness, Including Tongue	Height of Shoulder	Screw Dimensions				
	Width[2]		Length[3]	Projection				Diameter of Screw	Number of Screw	Threads per In.	Diameter of Head	Thickness of Head
	R_1	R	L	P	S	K	E	D			A	G
1/4	1/4	9/32	3/8	3/32	1/8	7/32	3/32	0.125	5	40	0.196	0.081
5/16	5/16	11/32	15/32	1/8	5/32	9/32	1/8	0.164	8	32	0.260	0.107
3/8	3/8	7/16	9/16	1/8	3/16	5/16	9/64	0.190	10	24	0.303	0.124
1/2	1/2	9/16	3/4	1/8	7/32	11/32	5/32	1/4	....	20	0.375	0.130
5/8	5/8	11/16	15/16	1/8	1/4	3/8	5/32	1/4	....	20	0.375	0.130
3/4	3/4	13/16	1 1/8	5/32	9/32	7/16	3/16	5/16	....	18	0.438	0.150
1	1	1 1/16	1 1/2	7/32	11/32	9/16	1/4	3/8	....	16	0.500	0.170
1 1/4	1 1/4	1 5/16	1 7/8	1/4	3/8	5/8	9/32	3/8	....	16	0.500	0.170
1 1/2	1 1/2	1 9/16	2 1/4	5/16	7/16	3/4	11/32	1/2	....	13	0.625	0.210

[1] T-slot dimensions will be found in Table 1.
[2] The "width of tongue" in the above table includes the recognized secondary standard having widths R_1 the same as the diameter of the corresponding T-bolt. This is to provide for the use, during the transition period, of this standard on many machine-tools where it is already established.
[3] The "length of tongue" can be varied to suit conditions.

TABLE 8—PROPOSED DIMENSIONS FOR COMBINATION REVERSIBLE TONGUES FOR T-SLOTS[1] WITH TWO SIZES OF T-BOLTS AND FOR TWO WIDTHS OF T-SLOTS WITH THE SAME SIZE T-BOLT

Diameter of T-Bolt[2]		Tongue Dimensions						Depth of Seat	Total Thickness Including Tongue	Height of Shoulder	Thickness of Land	Screw Dimensions				
Small	Large	Width[2]				Length[3]	Projection					Diameter of Screw	Number of Screw	Threads per In.	Diameter of Head	Thickness of Head
		R_1	R	R_3	R_2	L	P	S	K	E	C	D			A	G
1/4	5/16	9/32	11/32	5/16	1/4	15/32	1/8	3/16	5/16	1/8	1/16	0.164	8	32	0.260	0.107
5/16	3/8	11/32	7/16	3/8	5/16	9/16	1/8	7/32	11/32	9/64	1/16	0.190	10	24	0.303	0.124
3/8	1/2	7/16	9/16	1/2	3/8	3/4	1/8	1/4	3/8	5/32	1/16	0.250	....	20	0.375	0.130
1/2	5/8	9/16	11/16	5/8	1/2	15/16	1/8	9/32	13/32	5/32	3/32	1/4	....	20	0.375	0.130
5/8	3/4	11/16	13/16	3/4	5/8	1 1/8	5/32	5/16	15/32	3/16	3/32	5/16	....	18	0.438	0.150
3/4	1	13/16	1 1/16	1	3/4	1 1/2	7/32	3/8	19/32	1/4	3/32	3/8	....	16	0.500	0.170
1	1 1/4	1 1/16	1 5/16	1 1/4	1	1 7/8	1/4	7/16	11/16	9/32	1/8	3/8	....	16	0.500	0.170
1 1/4	1 1/2	1 5/16	1 9/16	1 1/2	1 1/4	2 1/4	5/16	1/2	13/16	11/32	1/8	1/2	....	13	0.625	0.210

[1] T-slot dimensions will be found in Table 1.
[2] The above table provides for a series of tongues so that a pair of tongues can be used with fixtures to machines of different sizes, using a different size T-bolt, and can also be used with fixtures on different machines having two widths of T-slots for the same size T-bolt.
[3] The "length of tongue" can be varied to suit conditions.

REVISION OF CLUTCH-FACING SIZES

Subdivision Reports Revised and Extended List for Single-Plate Clutches

At the time the present S.A.E. Recommended Practice for Single-Plate Clutch Facings was adopted, it was hoped that the sizes approved would come into general practice. However, a recent survey indicated the desirability of revising the specification. A Subdivision of the Transmission Division was therefore appointed, the members thereof being D. E. Gamble, of the Borg & Beck Co., chairman; J. J. Morris, of the Rockford Drilling Machine Co.; and E. E. Wemp, of the Long Mfg. Co. At a meeting of the Subdivision, which was held in Detroit on Nov. 11, 1926, prior to the meeting of the Transmission Division, current practice was reviewed and the following revision proposed:

Nominal Size, In.	Diameters, In.	
	Woven Type	Molded Type
8	7⅞ × 5⅜	7¾ × 5½
9	8⅞ × 6⅛	8¾ × 5¾
10	9⅞ × 6¾	9¾ × 6¼
11	10⅞ × 6¾	10¾ × 6½
12	11⅞ × 7¼	11¾ × 6½

The thickness of the woven type of facing shall be 1/8 or 5/32 ± 0.005 in.

The thickness of the molded type of facing shall be 9/64 ± 0.005 in.

The Subdivision's report was approved unanimously at the meeting of the Transmission Division and is submitted with the recommendation that it be adopted as a revision of the present S.A.E. Recommended Practice given on p. E19 of the S.A.E. Handbook.

Transmission Division Personnel

The personnel of the Transmission Division is as follows:

S. O. White, *Chairman*	Warner Gear Co.
P. L. Tenney, *Vice-Chairman*	General Motors Corporation
H. E. Blood	Detroit Gear & Machine Co.
A. C. Bryan	Durston Gear Corporation
L. C. Fuller	Fuller & Sons Mfg. Co.
D. E. Gamble	Borg & Beck Co.
A. A. Gloetzner	Covert Gear Co., Inc.
W. G. Hawley	American LaFrance Fire Engine Co.
D. F. Myers	Service Motors, Inc.
H. W. Sweet	Brown-Lipe Gear Co.
C. E. Swenson	Mechanics Machine Co.
H. T. Woolson	Chrysler Corporation

CLUTCH NOMENCLATURE REVISED

Transmission Division Recommends Slight Changes in the Present Standard

The Transmission Division at its meeting in Detroit on Nov. 11, 1926, voted to recommend that the nomenclature on p. K11 of the S. A. E. Handbook be revised by changing the term "Clutch shaft bearing (not in transmission case)" to read "Clutch pilot bearing (in flywheel)" under Group 1 of Division X, and to substitute the word "yoke" for "fork" in Group 2 of Division X.

TRACTOR MEETING AROUSES INTEREST

Sessions under Auspices of Agricultural Engineers and of Society

Society members interested in the industrial applications of the tractor assembled in the Crystal Room of the Hotel Sherman on Dec. 3, the last day of the joint meeting of the American Society of Agricultural Engineers and the Society which was held on Dec. 1, 2 and 3. The sessions on the first 2 days, which were devoted to agricultural problems, were under the auspices of the American Society of Agricultural Engineers; those on the last day being under the auspices of the Society.

At the morning session on Dec. 3, at which O. B. Zimmerman was chairman, Ralph H. Sherry, consulting metallurgical and industrial engineer presented an interesting paper on Metallurgy and Dr. Gustav Egloff, of the Universal Oil Products Co., presented a paper on Anti-knock Gasolines. The papers and the discussion thereon are covered in full in the following pages.

At the afternoon session papers were presented by William Parrish, of the International Harvester Corporation of America, Inc., on Industrial Application of Tractors and by T. Warren Allen, chief of the division of control of the Bureau of Public Roads, on Fields and Requirements for Automotive Equipment in Highway Building. These papers were printed in the December issue of THE JOURNAL. The discussion of these papers is given in full in the following pages. O. W. Young acted as chairman of the session.

The sessions on Dec. 1 and 2 were devoted largely to the discussion of the Combine, and its application throughout the United States, a subject that the meeting proved is of tremendous interest to the agricultural industry at the present time. The Combine, which is so-called because it *combines* in one operation both harvesting and threshing, is best adapted to humid areas, but where its use is possible it has cut the cost of labor four-fifths and has made it possible for the farmer to harvest his crops without calling upon bankers for financing. The development of the Combine is probably the most important agricultural development of recent years, and the interest stirred up by the discussion of the various phases of its use indicated clearly the reason for the large attendance at the sessions of the American Society of Agricultural Engineers. One of the most interesting thoughts brought out at the sessions that were held under the auspices of that Society was expressed by Thomas D. Campbell, president of the Campbell Farming Corporation, operating a 100,000-acre farm. Mr. Campbell felt that agriculture now represents the greatest possible field for inventive ability and that "mechanical agricultural engineering" should develop in the next 10 years into one of the most important branches of mechanical engineering.

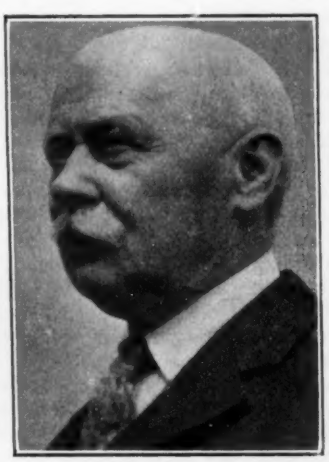
TOPICS OF AGRICULTURAL ENGINEERS

Harvester-Thresher of First Interest—Traction and Fuels Discussed

Capabilities and limitations of the Combine, or harvester-thresher, provided the subject of the first day's sessions of the farm power and machinery division of the American Society of Agricultural Engineers on Dec. 1 at the Tractor Meeting in Chicago, which was attended by the largest body of men ever gathered together for the sole purpose of considering the engineering aspects of mechanical equipment for the farm. As such, it was significant of the extent to which agriculture is coming to rely upon engineering.

The session opened with a paper on The Spread of the Combine, by F. A. Wirt, of the J. I. Case Threshing Machine Co., which dealt with the extension of the use of this comparatively new harvesting machine from the winter-wheat section into Illinois for harvesting soy beans and then through the corn belt and other areas for harvesting a variety of crops. Figures were given to show the great saving effected in labor by the Combine, and Mr. Wirt said that in his judgment the investment in a 16-ft. Combine for use in competition with other methods of harvesting on acreages of 160 acres or more is justified. He believes


O. W. Sjogren, Chairman


J. F. Max Patitz


O. W. Young


O. B. Zimmerman

THE TRACTOR MEETING COMMITTEE

that the driving of Combines by power take-off from the tractor is entirely feasible on reasonably level ground. Obstacles to successful and economical use of the Combine are weeds and other green vegetation, insufficiently ripened grain and the lodging of grain, which occurs to a larger extent when cutting is postponed until the grain is fit for operation of the machine.

A symposium of short papers on the same subject was presented after Mr. Wirt's paper by engineers of the Department of Agriculture and a number of agricultural colleges. C. D. Kinsman gave a progress report on the Department's investigations of harvesting with the combine. Robert H. Black, also of the Department, presented motion pictures showing the machine in operation and said that one feature of the last year's work was the building and trial of a special grain-thresher into which the disc type of dockage separator is incorporated as an integral part instead of being used as an attachment. Another new development is the forming of indentations in the surface of a rubber belt that operates at an angle of about 45 deg. instead of being located in the sides of flat discs that revolve in a vertical plane. This belt separator offers considerable promise as regards low cost and light weight.

Some Troubles and Means of Remedy

Papers on the work of the combine in different States and in various crops were given by R. C. Miller, of North Dakota; E. A. Hardy, of the University of Saskatchewan, Canada; Prof. E. W. McCuen, of Ohio State University; R. H. Wileman, of Purdue University; Prof. F. W. Duffee, of the University of Wisconsin; William Aitkenhead, of Purdue University; I. P. Blauser, of the University of Illinois, and J. M. Smith, of the University of Alberta, Canada.

These papers dealt with the practical use of the Combine rather than with engineering problems and design. Several of the speakers dwelt briefly on possible trouble due to moisture in threshed grains and means used to dry the grain in bins and in sacks. In the harvesting of soy beans some difficulty was encountered, said Mr. Blauser, where bull nettles were present, and under this condition the bar type of cylinder seemed more satisfactory. The best result was secured when the Combine was especially arranged to operate within 4 in. of the ground, and the beans are picked up better when the reel runs faster than usual in relation to the forward travel of the machine. In harvesting clover seed, he said, Combines fitted with their own engines gave better results than those driven from the tractor, because the forward movement of the outfit could be stopped, when choking or clogging was threatened, without interrupting the steady operation of the Combine. The same advantage could be secured, however, with the power-take-off drive if this could be taken from the engine through an independent clutch.

Latest engineering developments in low-cutting attachments for corn binders for control of the corn-borer infestation were described by C. O. Reed and Prof. E. W. McCuen, of Ohio State University, who reported some difficulty due to the tendency of the machine to pull the stalks due to too low an operating-speed, insufficient clearance, failure to cut low enough, and inadequacy of bull-wheel traction in one type.

A notable achievement in designing and building a combination corn picker, husker and stalk shredder was described. This was built by John Deere & Co. and the first machine is now at Ohio State University for testing. Professor McCuen asserted that the motor cultivator is the keystone of the corn-production arch and said that a four-row planter and four-row cultivator were used with excellent satisfaction in trials. In tractor work, lubrication time is reduced from 9.5 per cent of operating time to 3.9 per cent when the machine is fitted with lubricators of the contact type used in connection with a suitable high-pressure gun. Quick steering and short turning are important in relation to time consumed in turning at the end of the rows.

Under the subject, The Power Take-Off Uptodate, F. N. G. Kranich, of the Timken Roller Bearing Co., said that further engineering progress is dependent upon the builders of tractors and tractor-driven machines cooperating in a program of standardization of sizes, forms and locations of mating parts. He suggested that the committee which is to take up this problem consider two sizes of take-off shaft, one 1⅛ in. in diameter for motor cultivators and tractors of the two-plow type, and the other 1⅜ in. in diameter for use on larger tractors that transmit considerable power to Combines and other machines of similar power requirements.

Wheat Manufacture Shown in Motion Pictures

The showing of 3000 ft. of motion pictures entitled Manufacturing Wheat on a 100,000-Acre Farm Factory, by Thomas D. Campbell, of the Campbell Farming Corporation, of Hardin, Mont., was of popular interest on the second day of the meeting. The reel showed processions of a dozen or more tractors drawing gangs of plows; crawler-type tractors hauling trains of a dozen wagons loaded with 200 bu. of wheat each; the threshing of 4321 bu. in a 14-hr. day with an all-steel threshing machine; a tractor pulling four 10-ft. binders with their binding-heads removed and the last three machines provided with conveyors to deliver the cut material into a windrow, and a tractor-drawn combine with the platform raised and a hay-loader hitched behind to pick up the windrow and deliver it to the platform, whence it was carried into the cylinder and threshed in the usual way.

In a technical engineering paper on Field-Tractor Lug Studies, John W. Randolph, of the Alabama Polytechnic Institute, told of experiments conducted on Norfolk sand, which affords extremely poor traction, and said that the studies have resulted in the establishment of the principles of traction in such soils and the development of a mathematical formula by which the performance of a tractor wheel of specified dimensions, loading and lug equipment can be predicted with a high degree of accuracy. By the application of these principles, the available drawbar pull of a tractor has been increased 150 per cent over that obtainable with standard equipment and the tractor enabled to develop its rated power on the poor footing of Norfolk sand, he said.

Research Opportunities in Tractor Field

A look into the future was given by Prof. J. B. Davidson, of Iowa State University, in a paper on the Possibilities in Tractor Research. Speaking of the heat values and comparative cost of various kinds of fuel now used and likely to be used in the future, he said that the cheapest power does not come necessarily from the cheapest form of heat energy. Coal is the cheapest source of heat at present typical prices, as it gives about 2½ times as many heat units per cent of course as its nearest competitor, kerosene. Corn and hay give about two-thirds as much heat per cent as kerosene, and gasoline gives about one-half as much. Alcohol he deems worthy of consideration, while furfural, which is now produced at high cost and in small quantities chiefly from oat hulls, can be produced from a large variety of cellulose-bearing farm waste. Furfural merits consideration because of the possibility that it may ultimately be produced at costs which will enable it to compete with other fuels and also because it promises to be a valuable anti-knock material.

Although the thermal efficiency of the tractor engine is much higher than that of the automobile engine, said Professor Davidson, it is still far below that of the best stationary engine practice and offers room for substantial improvement, perhaps along the line of the constant compression engine. Another great opportunity for research lies in finding ways of reducing the power consumption due to traction, which consumes from 30 to 60 per cent of the power developed in practical field conditions. This suggests the question whether the present practice of running a tractor back and forth over soft ground is the best method of accomplishing field work.

WHAT GOOD ENGINEERING INVOLVES

Poor Material an Alibi for Failures Due to Design and Manufacturing Methods

Factors that must enter into the manufacture of any product that is to render good service were the theme of a paper on Engineering for Service in Materials, Design and Manufacture, which was presented by Ralph H. Sherry, consulting engineer of Chicago, at the session of the Tractor Meeting on the morning of Dec. 3, which was illustrated with a dozen lantern slides. The whole paper was an exposition of the thoughts embodied in the conclusion, in which Mr. Sherry said:

RALPH H. SHERRY

> Materials, design and manufacture go hand in hand. The manufacturer who recognizes this is seldom in trouble and seldom needs to buy the more expensive materials that he is told will stop trouble, and his manufacturing methods represent the lowest possible expense. Lack of knowledge of the more obscure causes of failure often is responsible for specifications that are much more strict than is necessary, for the choice of more expensive materials than are needed, for the rejection of materials that would give satisfactory service, and for time wasted in attacking the problem from the wrong angle, all of which unnecessarily increase the expense and are possible reasons for continued service problems. The successful manufacturer sets specifications as a guide but knows how to make satisfactory use of material that may not come within the specification range. His materials are selected, his designs made and his manufacturing methods established on a basis of experience and good judgment, and the result in service is his final guide.

CONDITIONS OF USE OUTGROW TESTS

The right materials to use are, in general, selected upon the basis of their composition and condition, and specifications are written accordingly. The usefulness of tests that provide the basis of these specifications may be outgrown as the conditions of use of a product change and the tests may no longer give protection, hence no test should be accepted as final until proved adequate by actual service results and should be discarded as soon as it is found of no practical value. High tensile-strength, which defines a certain physical condition, may not mean service strength; for example, said Mr. Sherry, a steering-knuckle of hardened tool-steel of high tensile-strength would not give long service, particularly in a tractor. Specifications may also cover manufacturing methods, such as heat-treatment and machine-shop operations. They can be used both as a guide to the supplier and as an alibi for trouble. In the final analysis, the service test is the one upon which dependence must be placed.

Metals, which are the principal materials with which the automotive engineer is concerned, are divided, said the speaker, usually into ferrous metals used for structural purposes and non-ferrous metals used for bearings. The latter are used for their non-friction, non-rusting and certain other characteristics. Methods have been developed recently for producing a very uniform bronze containing 70 per cent of copper, 19 or 25 per cent of lead and 11 or 5 per cent of tin and which can be remelted without segregation of the lead. Mr. Sherry gave typical analyses of several babbitt metals, which contain tin to give plastic qualities and copper and antimony to impart strength and wear-resistance, and showed a photomicrograph of a typical babbitt alloy which revealed cubic and needle-like crystals.

CARE NECESSARY WITH FERROUS METALS

The composition and characteristics of cast and wrought ferrous metals, including steel alloys, were dealt with at some length by the speaker, who said that when casting either ferrous or non-ferrous metals free flow of the poured metal into the mold is a basic requirement and gates and risers must be provided to allow for this or hard spots may result. The castings must be free from blow-holes and porosity, hence melting and pouring temperatures must be controlled.

Increased hardness and tensile-strength are imparted to steel by heat-treatment and also by the addition of alloying elements. Some alloys require more expensive handling and treatment than others of equal service qualities. Stainless steel, which contains 14 per cent or more of chromium, has entered into automotive construction comparatively recently to replace ordinary steel in places where corrosion is a cause of trouble. Because of the composition of the ore, steel contains small quantities of other elements, notably sulphur and phosphorous. The microscope reveals small inclusions of manganese sulphide or other materials in the steel, usually surrounded by areas of pure iron in streaks that follow the rolling lines, as in Fig. 1. In actual size the large band is about 0.040 in. long and 0.002 in. wide. All steel has a tendency toward lamination, particularly low-carbon low-chrome-nickel steel, as shown by the photomicrograph reproduced as Fig. 2. Tensile-strength is not a safe guide to strength of alloy steels to resist shock or alternating stress, but the increased hardness and tensile-strength are useful where resistance to tension, torsion or compression is desired.

DESIGN AND TREATMENT OFTEN CAUSE FAILURE

Improper design has perhaps been responsible for more failures than defective material, said Mr. Sherry. The effect of nicks or sharp corners is not well recognized. Sharp corners, holes drilled in dangerous places, minor kinks, and even tool marks sometimes result in breakage. Fillets are necessary when bending stresses are to be met, and the grinding-out of tool marks has been known to stop trouble. A crack sometimes starts at a sharp corner, forging-hammer marks or slight machining irregularities. Resulting breakages are attributed commonly to poor material and are responsible for the substitution of more expensive material or methods that do not solve the problem. Cracking of plow discs due to sharp corners in square holes for bolts near the center has been a common occurrence.

An example of failure due to error in manufacturing is that of the cutting teeth of mowers and reapers, which are

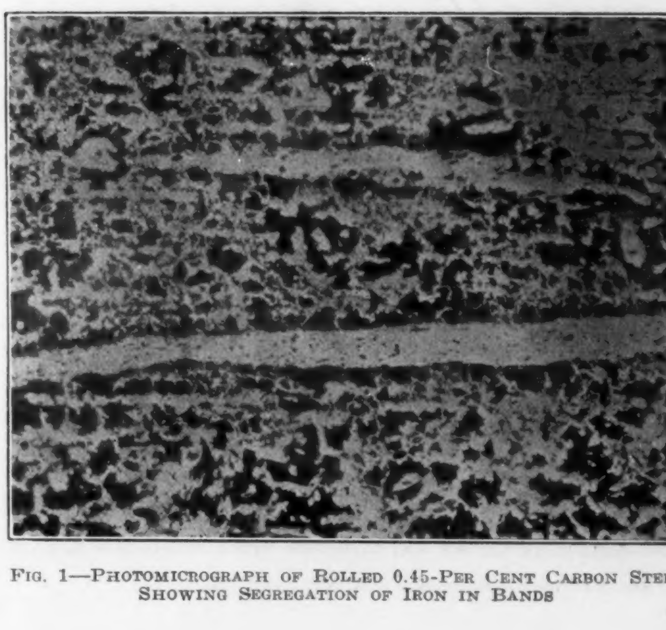


FIG. 1—PHOTOMICROGRAPH OF ROLLED 0.45-PER CENT CARBON STEEL SHOWING SEGREGATION OF IRON IN BANDS

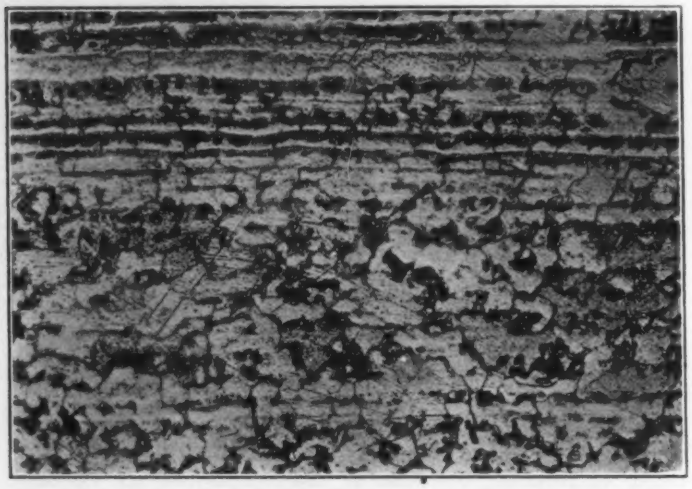


FIG. 2—LAMINATION IN LOW-CARBON LOW-CHROME-NICKEL STEEL DUE TO SEGREGATION OF CARBON-CONTAINING STRUCTURE

hardened on the cutting edge. If the greater part of the section is exposed to the heat because of irregularity in the castings that form the link of the chain that passes through the furnace, the hardening may extend all the way across and make the tooth sections too brittle.

The best materials in the world are not proof against failure due to errors in design and treatment that may be neglected because they are obscure. Excuses that are found in minor variations from specifications or analyses in the material may be satisfactory to the finder, but are ineffectual in removing the cause of trouble.

Replying to a question whether the double treatment of carburized gears is of any advantage, Mr. Sherry said that if it is necessary to refine the core the gear must be double treated, but the carburizing method is so thoroughly controlled that practically no necessity for the double treatment of gears exists; the same refining effect can be obtained by quenching direct from the box.

FUTURE SUPPLY OF MOTOR FUEL

Sources and Productive Capacity Analyzed; Adequate Future Supplies Predicted

Statistics to show that, based on the average quantity of motor fuel used per car in the United States in 1926, approximately 150,000,000,000 gal. per year would be required to supply a future world-potential of motor-fuel demand of 300,000,000 motor cars, were presented by Dr. Gustav Egloff, director of research of the Universal Oil Products Co., Chicago, in the paper on Motor Car and Motor-Fuel Potential which he read at the session of the Tractor Meeting held on the morning of Dec. 3. He asked where this great quantity of motor fuel is to come from and then analyzed the present and the probable sources of supply.

Mr. Egloff is an eminent petroleum technologist, who has devoted himself to the art of thermal decomposition of hydrocarbons and the "cracking" of petroleum oils. He has made numerous improvements in the cracking and refining of petroleum distillates, and the results of practically all of this work have been published in various important technical periodicals.

At present, said Dr. Egloff, the world-production of crude oil is approximately 1,000,000,000 bbl., from which is derived approximately 14,000,000,000 gal. of motor fuel by normal distillation and by the "cracking" process. In his opinion, this quantity of motor fuel could be more than doubled for the same quantity of crude oil by proper utilization of the cracking process. He believes that a shortage of motor fuel cannot occur for centuries to come when the supply of crude petroleum, tar-sands, coal, oil-shale, and wood-tars is given full consideration. He stated that South America apparently has a vast sea of oil; that Russia, Persia and Rumania have large oil-production with a vast quantity of potential oil in reserve; and that oil-findings in Canada, Africa, China, and other countries are indicated which may develop in a large way when economic demands justify it. He mentioned also the vast tar-sand deposits in Alberta, Canada, which could supply the present world-demand for motor fuel for more than 100 years by utilizing the cracking process.

THE MOTOR FUEL OF THE FUTURE

After discussing potential sources of motor-fuel supply, such as bituminous coal, oil-shales, wood-tars, and the like, the speaker said that the motor fuel of the future will, in all likelihood, be a blend of normal motor-fuel and cracked motor-fuel derived from coal-tar, oil-shale, wood-tar or a combination of these, in varying percentages. This motor fuel will yield double the mileage per gallon over that now being obtained in the motor-car of today. The specification will be for anti-knock and for ease of starting the engine, and the fuel will have no color, odor, or "doctor" or sulphur tests as criteria of its utility. He stated also that the urge of the present period is to produce motor fuel having high anti-knock characteristics in quantities sufficient to operate high-compression engines which must come on the market so that more mileage per gallon of fuel will be obtained.

It was also brought out by Dr. Egloff that the anti-knock properties of motor fuel are a function of the type of hydrocarbons present and of the percentage to which they enter into the composition. In all motor fuels four groups of hydrocarbons are present, these being the aromatic, the napthene, the unsaturated, and the paraffin groups. The reaction velocity of the different groups of hydrocarbons with air and with oxygen varies widely. Depending upon the percentage of the four groups of hydrocarbons present in a motor fuel, it may be entirely non-detonating or detonating to a high degree in a high-compression engine. The higher the percentage of aromatic, napthene and unsaturated hydrocarbons present, the better a motor fuel will be from a non-detonating viewpoint. Ease of starting is also a function of the type of hydrocarbons present in a motor fuel. The cracking reaction can be controlled so that the non-detonating hydrocarbons can be produced in the percentages desired; hence, they will lend themselves particularly to blending with straight-run motor-fuel with a view toward improving its burning qualities. The remainder of the paper was devoted to analyses of the different groups of hydrocarbons and their properties, to discussion of the advantages and disadvantages of the various processes in use and projected for the production of motor fuel and to the difficulties attendant upon the proper "treating" of motor fuels.

REMARKABLE NEW PROCESS[1]

Questions propounded during the progress of the discussion of the paper elicited much additional and valuable information from Dr. Egloff. One reply of his to a question relating to the hydrogenation of coal characterized the work of Prof. Friedrich Bergius, of Heidelberg, Germany, in this regard, as a technical triumph of the first water. The coal is ground into pieces not exceeding 2 mm. (0.08 in.), in diameter and is mixed with the heavy part of the oil from a prior operation to a pasty, thick mass. The pasty mass is pumped into a reaction chamber made of a gun forging, at a pressure of about 3000 lb. per sq. in. While the retort is being heated and the pasty mass is passing into the chamber, hydrogen gas is also pumped into the retort. The coal takes up approximately 5 per cent, by weight, of hydrogen, which liquefies more than 90 per cent of the coal, as well as some carbon and ash, the liquid being drawn off continuously through a valve at the far end of the retort. Approximately 5 per cent is gas, which is taken off from the top of the retort. While under pressure, the pasty mass is heated to a temperature of approximately 450 deg. cent. (842 deg. fahr.). The speaker said that Dr. Bergius claims a possible production of approximately 45 gal. of gasoline per ton of coal.

[1] See *National Petroleum News*, Nov. 24, 1926, p. 53.

INDUSTRIES USE MANY TRACTORS

Built First for Plowing, They Are Now Used for Road Work and Hauling

WILLIAM PARRISH

Surprise was created among his hearers by the statement made in his paper by William Parrish, of the International Harvester Corporation of America, at the session of the Tractor Meeting in Chicago on the afternoon of Dec. 3, that approximately 12 per cent of all tractors that have been produced have gone into industrial application. In the discussion he said that he had been informed by some of the caterpillar - tractor builders that 75 per cent of their production goes into industrial use, such as road construction and maintenance, logging, hauling, and so on. One company asserted that not more than 10 per cent of its tractors are used in agriculture. Last year 24 per cent of the Fordson tractors produced were supplied for other than agricultural purposes. Some of the industrial uses were described briefly by Mr. Parrish and illustrated with lantern slides. His paper was printed in full in THE JOURNAL for December, 1926.

O. B. Zimmerman, of the International Harvester Corporation, when called upon by Chairman O. W. Young to introduce the speaker to the meeting, mentioned that the tractor was first a simple machine designed for big plowing operations, then was extended to the pulverizing of the plowed soil and to cultivation, which required lighter machines. Later the use of tractors was extended into road operations, and designs had to be changed. The early agricultural tractor did not fulfill requirements, as it could be used a maximum of only 50 days a year. Tractors for cultivation can be used about 150 days. Changes in design that enable the tractor to be used for industrial purposes have given it a definite place in traction and extended its use over the entire year.

Following delivery of the paper, Chairman Young declared that he was impressed by Mr. Parrish's statement of the percentage of tractor production going into industrial fields, and with what this may mean commercially in times when agricultural sales do not run high.

Referring to the stump-sawing outfit shown in a slide, F. N. G. Kranich, of the Timken Roller Bearing Co., inquired if it is used in land-clearing operations. Mr. Parrish replied that it is a comparatively new unit designed for use in the pine woods, where turpentine is extracted from the stumps, but eventually, because of its utility and economy, may prove more useful than the stump-puller. Engineers who saw it in operation claim that a stump with a 4-ft. base can be removed in 3 min. When he learned that hard-headed lumbermen were behind the project, he decided that it must have considerable merit. It and the other special machines are provided with safety devices. When a hole-boring machine strikes a stone, the boring device stops and is withdrawn and started down in another place.

HIGHWAY-BUILDING EQUIPMENT

Opportunities Existent for Improved Motorized Road-Making Apparatus Cited

In his paper on Fields and Requirements for Automotive Equipment in Highway Building, presented at the session of the Tractor Meeting that was held on the afternoon of Dec. 3, T. Warren Allen, chief of the division of control, Bureau of Public Roads, City of Washington, summarized present practice and future possibilities relating to the use of automotive equipment for plowing and breaking up the earth, loading it into wheeled scrapers, hauling them to the fill, drawing and operating elevating graders, power excavating, trucking dirt from elevating graders and power-shovels to fills, and hauling paving materials to be placed on the subgrade. The paper was printed in full in the December, 1926, issue of THE JOURNAL.

RECENT DEVELOPMENTS

During the discussion of the paper, several new types of equipment were mentioned. Among them, a large-size scraper of 12-cu. yd. capacity that receives electric power from a generator mounted on the tractor that pulls the machine was described by A. E. Loder, of the Caterpillar Tractor Co. One of two motors on the scraper raises and lowers the cutting part; the other motor operates the telescoping bowl. Another type is a construction grader having a 14-ft. blade. Its forward end is carried by the tractor that pulls it and its rear end is supported by two wheels. In each of these machines the man who drives the tractor controls the entire equipment. Mr. Loder indicated that machines of the foregoing types are yet in the development stage. Wheel-type scrapers that are pulled by tractors and are equipped with mechanical means for handling the dirt after it is in the scraper so that loading is done by power received from the tractor were mentioned also.

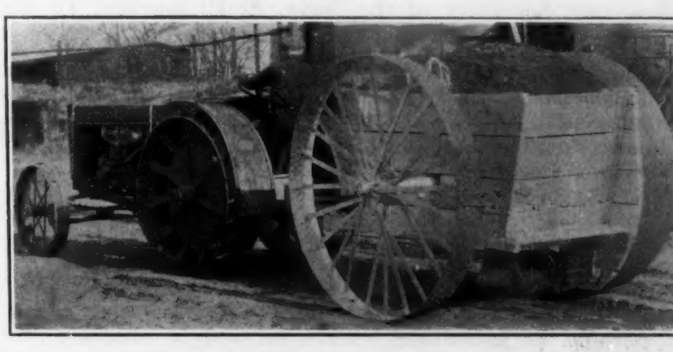
Replying to J. Otis Pierce, of the Brown-Lipe Gear Co., Syracuse, N. Y., who described a five-speed transmission recently developed for 2 to 2½-ton trucks that provides a low load-speed and an over-speed for saving time by speeding up return trips when empty, Mr. Allen said that such a transmission should be helpful to contractors, who must keep concrete mixers running continuously and who often encounter difficulty in doing this because they lack adequate hauling-equipment.

E. R. Wiggins, of French & Hecht, Davenport, Iowa, said that his company has devoted considerable study to the subject of rubber tires suitable for tractor use. He told of a recent 50 x 10-in. size, the largest size yet developed, now being used in dual form on tractors, and recommended further research by the industry as to allowable weight on rubber tires.

SPECIAL FEATURES

The use of rubber blocks on the treads of a caterpillar tractor was said to have given suprisingly good tractive results on roads that were not too "soupy" or muddy, the fact that a number of the rubber blocks on each tread are on the ground at the same time seeming to multiply the beneficial effects of using rubber contact with the road and to make the tractive effort more effective than is that of a rubber tire on a round wheel.

The development of so-called "back-up" hitches for use in



SELF-LOADING DUMP-CART HAULED BY A TRACTOR

Entirely Separate from the Tractor and Connected to It by a Linchpin, a Turn at Right Angles Can Be Made by the Outfit. The Cart Has a Capacity of 2 Cu. Yd. of Earth, and Loads Itself in 1 Min. by the Rolling Action of the Large Wheel Visible at the Right

short-haul grading was mentioned also, one such arrangement permitting the use of a small rotary-type scraper of greater than ordinary capacity that can be pulled directly to the dumping point, backed up or shoved back and forth from the ditch up onto the grade, if that is the direction in which the dirt is being handled, and used without turning the machine around.

An Interesting Self-Loading Dump-Cart

G. C. Andrews, of the Glide Machine Co., showed several photographs of a self-loading dump-cart of 2-cu. yd. capacity which is hauled by a tractor. The cart is a separate unit and is connected to the tractor by a linchpin. Thus connected, a turn at right angles can be made in any street or on any grade. The earth is elevated by the rolling action of the large wheel shown at the right, which has an annular space about 8 x 8 in. in cross-section. As the wheel rolls forward, the earth is carried up to a space above the box; there, the earth is removed by a scraper, thrown across and distributed over the surface of the box. The cart can be loaded to capacity in 1 min. The dirt is thrown into the opening at the base of the wheel by the plow or grader blade shown underneath, which can be set to any depth desired by the operator on the tractor. A spring release prevents injury to the machine when striking stones or other obstructions. After it is hauled to the desired location, the box is emptied through doors similar to those for ordinary dump-wagons for earth. The doors drop down and are controlled from the tractor by the operator.

Use of this cart reduces the expense of moving earth to that of the mere cost of transportation, since no expense is entailed for loading. In almost all types of earth removal, the self-loading feature reduces the cost of operation one-half. Considering an outfit of 10 team-wagons and an elevating grader, the expense of the elevating grader, averaging from $50 to $75 per day, is eliminated by the outfit shown. Since each unit of the type shown works independently, it is not necessary to use a unit of greater capacity than is required for each individual job.

NEWCOMB INADVERTENTLY PROMOTED

In the account of the Transportation and Service Meeting Banquet that appeared on p. 525 of the December, 1926, issue of The Journal, R. H. Newcomb, who acted as toastmaster, was referred to as vice-president of the New York, New Haven & Hartford Railroad Co., and president of the New England Transportation Co. Arthur P. Russell, who presented a paper at the Transportation and Service Meeting dealing with the Theory and Method of the New Haven Railroad's Highway Operation, holds these two offices, Mr. Newcomb being his assistant.

R. H. GRANT TO SPEAK AT DINNER

Assistant Secretary of the Navy, E. P. Warner, To Act as Toastmaster

Members of the Society will be pleased to know that the principal address at the Annual Dinner, on Jan. 13, at the Hotel Astor, New York City, will be made by R. H. Grant, vice-president of the Chevrolet Motor Co. His subject will be What Sells Motor-Cars.

Society members had the privilege of hearing Mr. Grant at the 1922 Service Meeting in Chicago. At that time Mr. Grant's subject was How the Engineer Can Help Business and his address, which was printed in full in the March, 1922, issue of The Journal, was keenly appreciated. Much that Mr. Grant said at the Service Meeting Dinner is still applicable to present-day conditions and if members will but read this address again they will, considering the subject decided upon for the 1927 Dinner, appreciate the importance of hearing Mr. Grant on Jan. 13.

Immediately following the dinner, President Lítle will call the Annual Society Business Meeting to order and Tellers of Election will submit their report on the election of officers for 1927. After the report is accepted the Business Meeting will be adjourned to Tuesday, Jan. 25, at Detroit. Mr. Lítle will then introduce the toastmaster, the Hon. E. P. Warner, Assistant Secretary of the Navy. Mr. Warner has been one of the Society's most active members during the last 10 years and all members will look forward to hearing him. J. H. Hunt, who will make his first appearance as President-Elect at the dinner, will give a short address.

A special table in the balcony directly opposite the speakers' table will be reserved for the ladies this year. This innovation has been made possible by the rebuilding of the balcony. Members who cannot be accommodated on the ballroom floor will be seated in the balcony, making it unnecessary to use the Laurel Room as in previous years and enabling everyone to hear the speakers.

Under the chairmanship of H. O. K. Meister the Dinner Committee is appointing a member to serve as host at each table. The names of these hosts will be indicated in the seating program to be given out at the Dinner. A hostess will also preside at each table reserved for the ladies.

The entertainment this year will include Wolfsie's orchestra, Maurice Garabrant at the organ and Frank Sherman, who will act as song leader. Several interesting novelties have been arranged by the Dinner Committee.

Over 800 reservations have been made for the Annual Dinner and based on previous records it is believed that the attendance will be the largest of any Annual Dinner.

METROPOLITAN SECTION DINNER

The dinner that has been given by the Metropolitan Section during New York Automobile Shows in recent years will be held again this year. The subject What's New at the Show? will be assigned to each speaker. The Metropolitan Section Dinner will be held at the Hotel Commodore on the evening of Monday, Jan. 10.

The list of speakers will include representatives of all the well-known automobile companies and at the close of the meeting Austin M. Wolf will summarize the statements made by the different speakers and indicate the general trend of engineering developments.

The Metropolitan Section has arranged a very interesting program of music, favors and professional entertainers which is expected to surpass that of any previous dinners.

ELEVEN TECHNICAL SESSIONS SCHEDULED

Alan Fenn of Sunbeam-Talbot-Darracq To Present Paper on English Light Car

Possibly the most interesting Session at the 1927 Annual Meeting will be that devoted to the discussion of the light car. Papers will be presented by Alan Fenn, of the Sunbeam-Talbot-Darracq Combine and Fabio Sergardi, of the Reo Motor Car Co. Mr. Fenn kindly accepted the invitation of the Meetings Committee, extended to him through W. O. Kennington, to come to this Country expressly to present a paper covering the English light car.

Mr. Fenn's first connection in the automotive industry was with the Simms Mfg. Co., which not only built cars but was responsible for pioneering the high-tension magneto in Great Britain. Later Mr. Fenn personally undertook the construction of passenger-cars, subsequently amalgamating with T. O. M. Sopwith. When Mr. Sopwith decided to take up flying exclusively, the partnership was dissolved and Mr. Fenn went with the Metallurgique Company which had the agency for the Metallurgique car in Great Britain.

During the war Mr. Fenn joined the Royal Naval Air Service, after being rejected by both the Army and Navy because of physical disability. He exercised responsible supervision in connection with aircraft engines, and later with complete aircraft. The plants under his supervision turned out over 6000 Sopwith airplanes. After the war, Mr. Fenn

ANNUAL MEETING PROGRAM

General Motors Building Jan. 25 to 28 Detroit

Tuesday, Jan. 25

9:00 a.m.—Registration

9:30 a.m.—Standards Committee Meeting

11:30 a.m.—Research Committee Meeting

12:30 p.m.—Luncheon

1:30 p.m.—Registration

2:00 p.m.—Training Session

Apprentice Training—H. A. Frommelt

Training at General Motors Institute of Technology—Albert Sobey, Flint Institute of Technology

2:00 p.m.—Body Session

Color Harmony—Arthur S. Allen, Ruxton Color Service

Motion Pictures of Hudson Body Plant—V. P. Rumely, Hudson Motor Car Co.

Recent Developments in the Weymann Body—Charles Weymann, Weymann American Body Corporation

Body Design—R. H. Dietrich, Dietrich, Inc.

8:00 p.m.—Business Session

Wednesday, Jan. 26

9:30 a.m.—Production Session

Hardness and Machinability Tests of Grey-Iron Castings—E. J. Lowry, Hickman, Williams & Co.

Chromium Plating—W. N. Phillips, General Motors Corporation

9:30 a.m.—Detonation Symposium

Petroleum Motor-Fuels, Their Detonation Characteristics—W. A. Gruse, S. P. Marley and D. P. Stevens, Mellon Institute of Industrial Research

Detonation Specifications of Fuels—Graham Edgar, Ethyl Gasoline Corporation

Study of Detonation and Detonation Control by the Ultraviolet Spectroscopy of the Flames of Motor-Fuels—G. L. Clark, Massachusetts Institute of Technology

Methods of Measuring Detonation—H. K. Cummings, Bureau of Standards

Causes and Effects of Detonation—Thomas Midgley, Jr., Thomas & Hochwalt Laboratories; Stanwood W. Sparrow, Studebaker Corporation of America; R. E. Wilson, Standard Oil Co. of Indiana; and T. G. Delbridge, Atlantic Refining Co.

12:30 p.m.—Luncheon

2:00 p.m.—Production Session

The Application of X-Rays in the Automotive Industry—G. L. Clark, Massachusetts Institute of Technology

Production Control—M. A. Lee, College of Engineering, Cornell University

2:00 p.m.—Research Session

Progress Report on Starting and Acceleration Tests—J. O. Eisinger, Bureau of Standards

Tendencies in Research at Engineering Colleges—A. A. Potter and G. A. Young, Purdue University

Report of Research Committee—Dr. H. C. Dickinson, Bureau of Standards

6:30 p.m.—Sections Committee Dinner Meeting

Thursday, Jan. 27

9:30 a.m.—Light-Car Session

The English Light Car—Alan Fenn, Sunbeam-Talbot-Darracq Combine

The Light Car—Fabio Sergardi, Reo Motor Car Co.

12:30 p.m.—Luncheon

2:00 p.m.—Mystery Session

5:30 p.m.—Meetings Committee Dinner Meeting

Friday, Jan. 28

9:30 a.m.—Transmission Session

Four-Speed Transmissions—C. A. Neracher and Harold Nutt, Durant Motors, Inc.

The Constantinesco Torque Converter—R. K. Jack.

12:30 p.m.—Luncheon

2:00 p.m.—Chassis Session

Rubber Spring Mountings—Walter C. Keys, United States Rubber Co.

Pitch, Toe-In and Caster—K. L. Herrmann, Studebaker Corporation of America

Why Does a Car Pivot?—Johannes Plum, Royal Danish Legation

9:00 p.m.—Carnival at Oriole Terrace

returned to England as a director of the Sopwith Aviation Co., and in 1920 joined the Sunbeam-Talbot-Darracq Combine, his work being largely as special liaison officer between the Board and subsidiary companies. In 1921 he was made General Manager and Director of Clement Talbot, Ltd., one of the subsidiary companies, and in 1926 he was transferred to the group organization of the Combine.

During the 5 years Clement Talbot, Ltd., was under Mr. Fenn's control the company turned out what was undoubtedly one of the most highly advanced and refined light cars produced in Europe and one, moreover, that has exercised a great influence upon light cars in general. Mr. Fenn is a member of the Council of the Society of Motor Manufacturers & Traders, the association of manufacturers that represents the automotive industry in the British Isles.

The complete program for the Annual Meeting is given on the preceding page, 11 sessions being scheduled. The Meetings Committee has made an effort to avoid any conflict of sessions and committee meetings that would make it impossible for members active in Society work to attend all meetings in which they are interested.

Four-Speed Transmissions

In view of the general interest in four-speed transmissions, the paper by C. A. Neracher and Harold Nutt of the Durant Motor Co. should prove of special interest. This paper will deal exhaustively with a type of four-speed transmission designed for quiet operation in third gear, fourth gear being direct, and ease of control. The paper will be followed by written discussion to be submitted by representatives of several passenger-car and transmission manufacturers. Following this discussion, R. K. Jack will present a paper on the Constantinesco Torque Converter. Through Mr. Jack the Meetings Committee invited Mr. Constantinesco to present a paper at the Annual Meeting on the design and operation of the mechanism, but as he could not come to this Country at this time, arrangements were made at the suggestion of the Meetings Committee, for Mr. Jack to prepare the paper. In view of the general appeal of this subject, it is believed that Mr. Jack's paper will be of extreme interest.

Wheel Shimmy Being Studied

K. L. Herrmann, of the Studebaker Corporation of America, who is preparing a paper on Pitch, Toe-In and Caster, has recently stated that the matter of shimmy has been in the past shrouded in mystery to a certain extent, but that after studying conditions as a result of being asked to prepare the paper by the Meetings Committee, he finds that the mysterious thing about shimmy is why any cars are able to run without shimmying. Mr. Herrmann's paper is bound to be of great value and the discussion following should do considerable to resolve the problem of shimmy into its fundamental elements.

Johannes Plum will present a paper at this session on Why Does a Car Pivot? Mr. Plum has a definite theory covering pivoting which definitely contradicts current opinion. His analysis has been confirmed by the Bureau of Standards and should prove of great interest to every engineer who has made a study of this subject.

Walter C. Keys, engineer of the automotive division of the United States Rubber Co., will present a paper on Rubber Spring Mountings at this session. Mr. Keys has been interested in spring-suspension for many years, his first paper on riding qualities, based on research carried on while at the Cadillac Motor Car Co., appearing in the December, 1917, issue of The Journal.

Allen, Rumely, Weymann, and Dietrich

Although Arthur S. Allen, of the Ruxton Color Service, has no business connection with the automotive industry, his work on color harmony in connection with containers and advertising has established a national reputation for his company which manufactures printing inks. His ideas, however, are entirely applicable to the painting of passenger-cars and it is with much pleasure that the Meetings Committee received his acceptance of its invitation to present a paper at Detroit. Mr. Allen was closely associated with A. E. O. Munsell whose system has been used as a basis for Mr. Allen's work. His paper will be accompanied by exhibits of various color effects and he has also consented to exhibit several enlarged photographs of cars showing good and bad color combinations.

Following Mr. Allen, V. P. Rumely, of the Hudson Motor Car Co., will show motion pictures taken in the new body plant of his company, and will point out during the presentation the various interesting processes and equipment. Short papers will also be presented by Charles E. Weymann, of the Weymann American Body Corporation, and R. H. Dietrich, of Dietrich, Inc. The Society is fortunate in being able to hear Mr. Weymann, as it is only a business trip to this Country at this time that makes it possible for him to attend the Detroit Meeting.

Complete details of all the sessions will be given in the *Meetings Bulletin* to be mailed to the members on Jan. 13.

Discussion Important

Each session at the Annual Meeting is being arranged so that the presentation of the papers will take less than half the allotted time of 3 hr., thus leaving 1½ hr. for discussion. The most valuable discussion is generally prepared beforehand, and arrangements are consequently being made to send mimeographed copies of the papers to all members wishing to discuss them.

This year there will be a breakfast or luncheon before each session for the chairman and speakers for the purpose of going over the session program and becoming acquainted. Members submitting written discussion in advance of the sessions will be invited to the breakfast or luncheon conferences in order that they may have an opportunity to meet the speakers beforehand.

DELIRIOUSLY AUTOMOTIVE

Carnival at Oriole Terrace, Detroit, Jan. 28, To Exceed All Previous Records

The Carnival Committee, under the chairmanship of E. V. Rippingille, has plans under way for what is expected to be the most successful and entertaining Annual Carnival ever staged by the Society. Since last year's Carnival, Oriole Terrace has been lavishly re-decorated, and when trimmed for the Carnival in a manner characterized by Mr. Rippingille as "deliriously automotive," it is believed that the Terrace will take on an atmosphere that will rival many a New Orleans Mardi Gras. It is really impossible to describe the Carnival adequately, but the special *Meetings Bulletin* mailed to the members of the Society by the Carnival Committee is an attempt to picture this unique event.

The Committee responsible for the 1927 Carnival consists of E. V. Rippingille, chairman, Fred A. Cornell, H. N. Davock, H. T. Ewald, W. R. Flannery, L. C. Hill, W. C. Keys, F. W. Marschner, P. N. Overman, M. P. Rumney, George Stone, and W. R. Strickland.

SECTIONS MEETINGS IN DECEMBER

UP-IN-THE-AIR EXPERIENCES RELATED

Aviators' Thrills Described and Flight Pictures Shown to Detroit Section

Motion pictures of the Detroit Arctic Expedition, of the Mount Clemens to Miami flight and of airplane landing on ice and taking-off therefrom made more vivid the descriptions of the varied thrills that aviators experience which were given by the speakers at the meeting of the Detroit Section that was held on Dec. 16. Major T. J. Lanphier, commandant of Selfridge Field and pilot on the Detroit Arctic Expedition, related experiences with airplanes, and Lieut. C. D. Williams, engineer for the Aircraft Development Corporation, Detroit, reviewed those incident to flight in free balloons. L. M. Woolson, aeronautical research engineer for the Packard Motor Car Co., Detroit, was chairman.

Major Lanphier said in part that the whole subject of pursuit tactics with airplanes is based upon the equipment used. The pilot invents his tactics as he flies, according to his need, and the tactics change to meet the demands of new or of different equipment. He remarked that the present type of pursuit airplane, which has a speed of about 160 m.p.h., is not as good as is needed and asked that engineers try to develop a machine capable of a speed of 200 m.p.h. A pursuit airplane must climb fast and be able to dive without demolishing its wings. Pilots of pursuit airplanes must at all times be aggressive. They have no defensive tactics and can shoot only from the front end. If attacked, they must turn if necessary before they can defend themselves by shooting from the front. Pilots must be reckless to a degree, but they must not be too reckless, because the machines of a squadron must keep together. Difficulty was experienced during the World War with pilots who would leave the squadron and fight alone. The main object is to rise above the enemy. As to flight in thick weather, the speaker said that the object is to climb above the clouds, and then gave instances of the extreme difficulties of doing so. Supplementing Major Lanphier's address, Major Brainard, commander-in-chief of the Marine Flying Corps, told of difficulties he had encountered while flying through fog in the vicinity of San Francisco and emphasized the dangers of flight in fog.

Free Ballooning

In his address, Lieutenant Williams said that, in a free balloon, only after finding the direction of the winds at different altitudes can the direction and destination of the balloon be determined. Under ordinary conditions, a free balloon can be steered remarkably well by taking advantage of the varying direction of the wind at different levels. For this reason a balloonist needs to study meteorology. Persons riding in the basket of a balloon have no sense of motion, even at a speed of 40 or 45 m.p.h., because the entire equipment moves as a part or particle of the atmosphere. In free-balloon distance-racing, everything possible is done to save gas and ballast so as to be enabled to maintain control of altitude for the maximum time. Experience is needed in landing a balloon without injury to the passengers. When traveling at a speed of about 35 m.p.h., the gas-valve cord is pulled when the balloon is about 50 ft. above the ground.

Thrills due to various unexpected causes were related, among them being the proclivities of rural would-be humorists to use balloons as targets and to shoot at them with rifles. After landing a balloon safely that had become heavily laden with snow, one aviator thoughtlessly dumped the snow off while handling the bag to retrieve it and was privileged to see his balloon soar away and vanish, not to be found again for more than a month. At great altitude, where the density of the air was only half that at ground level, one balloonist closed the valve to prevent the escape of gas, his mistaken idea being in some manner connected with equilization of pressure. The balloon burst, but the speaker neglected to mention what became of the balloonist.

Lieutenant Williams said that balloon racing is somewhat like playing a game of chess on a board that has squares as large as counties or states. Expert use of weather forecasts must be made to win a race. Radio reports of the weather are of great assistance.

SERVICE, OPERATION, MAINTENANCE

Meeting of Metropolitan Section with Other Bodies Breaks Attendance Records

Service, operation, maintenance, three topics of particular interest to the automotive industry, were explained, illustrated and discussed at a joint meeting of the Metropolitan Section with the Automotive Service Association, the Motor Truck Association of America and the Motor Truck Maintenance Club, at the regular monthly meeting of the Section at the Woodstock Hotel, New York City, on Dec. 16.

The Fundamentals of Automotive Service were described by T. L. Preble, regional service manager of the White Co.; motion pictures depicting the Development and Operation of Automotive Labor Saving Equipment were shown by John Stilwell, executive assistant and formerly superintendent of transportation of the Consolidated Gas Co. of New York; and an active discussion, led by J. F. Winchester, superintendent of motor equipment, Standard Oil Co. of New Jersey, was participated in by T. C. Smith, engineer, American Telephone & Telegraph Co.; R. T. Benham, service manager, Studebaker Corporation of America; Nat Mallouf, president of the Mallouf Haulage & Maintenance Co.; B. K. Rhoads, assistant general superintendent of motor-vehicle equipment, American Railway Express Co.; Ralph Werner, automotive engineer, C. A. Peirce, chief engineer, Diamond T Motor Car Co., Chicago; W. G. Gow, general service manager, Studebaker Sales Co., Newark, N. J.; Arthur Dunn, president, Anti-Stall, Inc.; and David Beecroft, of the Chilton-Class Journal Co.

The efforts of Chairman F. K. Glynn to arouse interest in the meetings and to increase the membership of the Section were rewarded with a record attendance, 273 members and their guests enjoying the customary dinner that preceded the meeting, while 50 members of the Motor Truck Association were dining and holding a special meeting in an adja-

T. L. Preble John Stilwell
Authors of the Two Papers Presented at the December Meeting of the Metropolitan Section

cent room. This number has been exceeded only by the meeting at the Hotel Commodore last January during the week of the Automobile Show. In opening the meeting Chairman Glynn introduced President J. W. Florida who in turn introduced Mr. Preble. President Joseph Husson of the Motor Truck Association and President John McLachlan of the Motor Truck Maintenance Club performed like services for Messrs. Stilwell and Winchester, respectively.

Fundamentals of Automotive Service

Mr. Preble called attention to the fact that more than 20,000,000 motor vehicles are registered in the United States and comprise 81 per cent of the entire registration of the world; and that the servicing of these vehicles is divided among a total of 79,000 repair-shops, 48,000 of which are maintained by car dealers and 31,000 are independent. The force necessary to perform the repairs consists of more than 480,000 employes. He stressed the point that motor-trucks are no longer peddled by old-time methods of salesmanship but that motor transportation must meet the buyer's needs and be capable of rendering efficient service. The service should include both the service built into the product and the organized service extended by the seller to the buyer. Analysis of the needs of the buyer should include the determining of the average and the worst operating-conditions and the application to such conditions of units adequately powered and equipped with the lowest gear-ratios necessary to meet the worst conditions.

The servicing of "orphan" trucks is expensive and unsatisfactory, continued Mr. Preble; inaccessibility of parts, and the requirement of special tools, are expensive; interchangeability of parts obviates excessive parts inventories; the finishing of various parts and the proper balancing of rotating or reciprocating parts has obvious significance from a maintenance standpoint. The importance of standardization of equipment and the keeping of cost data were also emphasized.

Service, said Mr. Preble, must go hand in hand with salesmanship and comprehends both helpfulness preceding a sale and continued interest after the product has been delivered. Its specific functions include (*a*) the manufacture and distribution of parts, (*b*) mechanical repair-work, (*c*) direct cooperation with the sales department, (*d*) direct cooperation with owners and prospects, and (*e*) cooperation with the manufacturing organization with a view to improving the product and assisting in adapting the product to customers' requirements. Proper parts service policies, in the opinion of Mr. Preble, on the part of the sellers of motor-vehicles should be founded on the proposition that the manufacturer or distributor should "hold the bag," and fleet-owners should take advantage of such a policy.

In conclusion, Mr. Preble asserted that a close adherence to specifications by manufacturers and more conscientious avoidance—of the common evils of overloading, overspeeding, lack of lubrication, and deferred maintenance represent a definite obligation, the fulfillment of which is highly profitable to owner and manufacturer alike. It cannot be too strongly urged that owners should make greater use of the facilities and cooperation placed at their disposal by manufacturers and, by the same token, that manufacturers must, in the best interest of the automotive industry, better fit themselves to carry out the additional obligations of service thus imposed. Constructive criticism between owner and manufacturer is certainly beneficial.

The Applications of Automotive Equipment

The motion pictures presented by Mr. Stilwell vividly portrayed the savings of time and labor made possible by the use of various types of automotive machinery. Among the applications shown were those involving the digging of post holes, the setting of the poles in the holes, the handling of a reel that weighed 4 tons, the drawing of three strands of cable through 400-ft. ducts at the same time, the loading and unloading of transformers, hoisting men into position for repairing street-lamps, pumping the condensation out of man-holes by a centrifugal pump, drilling holes in pavements with compressed-air, and reviving, with a pulmotor, a man who has been overcome by carbon monoxide gas.

TIRE'S RELATION TO ROAD AND VEHICLE

The Tire and Its Relation to the Road and the Vehicle and the various problems that arise in connection with the valves in balloon tires were described and discussed at the regular monthly meeting of the Northern California Section held at the Athens Athletic Club, Oakland, Cal., on Dec. 9. K. D. Smith, development engineer for the Firestone Tire & Rubber Co., Akron, Ohio, and Mr. Herger, Pacific coast manager of A. Schrader's Son, Inc., Brooklyn, N. Y., were the speakers. The membership committee through Secretary W. S. Stowell announced that 10 new applicants for membership in the Section had been received.

DEFECTS FOUND BY RADIOGRAPH

Use of X-Ray on Castings at Watertown Arsenal Told to the Detroit Section

From 75 to 90 per cent of the defects in plain carbon-steel castings made by representative foundries are due to mistakes that occur on the molding floor or in designs and most of which are preventable, asserted Dr. H. H. Lester, of the Watertown Arsenal, Watertown, Mass., in an address on The X-Ray Examination of Metals, at the Dec. 2 meeting of the Detroit Section. A prominent man in the metal industry told him, Dr. Lester said, that if the X-ray method could point the way to the elimination of preventable defects, the sale of steel castings could be increased by a factor of 5 at least.

The X-ray apparatus is used at the Arsenal, first, to make photographs of details of cross structure in the metal and, second, to examine the microstructure in steel. The former is called radiography and the latter X-ray metallography. By using X-rays in certain ways photographs can be obtained that make it possible to tell the relative position of different kinds of atoms in the material and how the microstructures behave individually when stress is applied.

The physical principles on which the apparatus operates were explained by a diagram and the speaker said that the short waves of the X-ray can penetrate 3½ in. of steel in 30 min. More than a dozen lantern slides showing defects revealed by the apparatus were displayed and explained. Several of the steel castings revealed cavities caused in some cases by reducing gas, in others by water vapor due to green sand in the mold and in others to shrinkage of the metal in solidifying. Risers, or enlarged portions of the casting, which are supposed to supply molten metal to compensate for shrinkage, do not always function as intended, said Dr. Lester, as the riser is sometimes too small. Sometimes the cavity does not show on the outside.

Cracks Avoided by Removing Cause

A number of other slides revealed cracks. Visible cracks usually are chipped-out in the foundry and welded, but often the cracks are concealed and again, in chipping, the chipper does not always get to the bottom of the cracks and the welding, even when well done, is often an incomplete remedy. At the Arsenal, welding is not done at all; instead, the cause of the cracks is determined and an attempt is made to eliminate it. In many instances cracks that did not appear either on the outside or inside were found by the X-ray in the fillets of pipe fittings that had a heavy flange cast on the top section. To avoid these it was decided to cast the flanges separately.

Defects obviously due to faulty design were found on five or six different occasions. The speaker emphasized that in the case of all castings a study should be made by the design engineers of the difficulties that the foundryman has to overcome in the pouring of metal into molds.

Regarding the extent to which the X-ray method can be used in routine testing, Dr. Lester said that where the cost of the service into which the casting is to go exceeds the

cost of the test, it would be foolish not to use the test. The experience at the Arsenal has been that the cost per pound of manufactured steel has been going down steadily although the volume of production has also been decreasing. This is due to the smaller number of rejections on both the important and the unimportant castings.

Replying to questions, Dr. Lester said that the cost of the apparatus is about $10,000; a radiographic unit has been established at Ann Arbor; the Bureau of Standards bought one about a year ago and has it nearly ready for use; the rule at Watertown Arsenal is to X-ray every new casting; other metals than steel can be X-rayed; the X-ray tube room at the Arsenal is lined with 12 tons of lead to protect the operator, who looks through a periscope, and that the fluoroscope might be useful for examining small automobile parts but will not penetrate metal more than ½ or ¾ in. thick.

WORK OF NEW AERONAUTICAL BUREAU

Patents and Designs Board's Duties Outlined for Chicago Section

Photograph from Wide World Photos
HON. W. P. MACCRACKEN, JR.

An exposition of the reasons underlying the creation of the new Patents and Designs Board for Aeronautics was made by Assistant Secretary of Commerce William P. MacCracken, Jr., a member of the Board, at the meeting of the Chicago Section that was held on Dec. 14 at the headquarters of the Western Society of Engineers, Chicago. About 45 members and guests were present at the informal supper provided in the Engineers' Club, and this number was increased to 70 at the opening of the meeting. O. W. Young, assistant sales manager in charge of tractor bearing sales for the Hyatt Roller Bearing Co., Chicago, was chairman.

The Patent and Designs Board for Aeronautics is composed of three assistant secretaries for aeronautics representing the War, the Navy and the Commerce Departments; F. Trubee Davison, Edward P. Warner and William P. MacCracken, Jr., respectively, are the present representatives. Secretary MacCracken said that, according to law, all designs that are to be submitted to the Board shall first be passed upon by the National Advisory Committee for Aeronautics. A paid staff passes the designs and sends reports on them to the Board. The duty of the Board is to consider all designs pertaining to the development of aeronautics which are submitted by individuals and corporations and to decide the possible worth of such designs to the Government. The speaker then explained in detail the actual functioning of the Board and cited numerous instances of actual experiences with inventors and others. He discussed also some of the successfully operated commercial air-lines and mentioned instances indicative that great progress in commercial aviation already has been made. Since the problems connected with flying through fog are among the most difficult at present existing, neon-gas lights and other devices for penetrating fog were enumerated and comments were made upon their efficacy.

SUCCESSFUL COMMERCIAL OPERATION

In the course of the discussion following the address, Chairman Young quoted interesting statistics relating to a commercial air-line between Salt Lake City and Los Angeles that already is self-sustaining after having been in operation for less than 1 year. Its airplanes have flown approximately 300,000 miles since April 17, 1926. During 500 trips over this 600-mile airway, on only three occasions was any mechanical difficulty encountered in flight. These difficulties were minor and only one occasioned delay. More than 100 passengers were carried between May 23, when passenger service was inaugurated, and the occurrence of unsettled weather in the fall. In air-mail volume, this line carried 20 per cent of the Nation's daily dispatch.

Secretary MacCracken estimated that air-transport operators today are making about 65 per cent of what they ought to earn to show a fair return on their investment, despite the fact that they are carrying less than 50 per cent of their capacity loads. With increased loading, with discoveries as to possible economies in the cost of operation that will result from greater experience, with possibilities of doubling the service by flights between terminals at night as well as by day, all possible of accomplishment with practically the same amount of overhead cost, the financial margin of safety will become greater and satisfactory earnings will become apparent.

STEEL AND FLEXIBLE BODIES ARGUED

Important Development of Fabric Type Predicted at the Indiana Section

Great interest in body construction was displayed in discussion at the Dec. 9 meeting of the Indiana Section at the Hotel Severin in Indianapolis following an address on steel body-construction delivered by L. E. Ruehlmann, of the Edward G. Budd Mfg. Co., and one on the flexible type of body, given by H. Steinbrugge, vice-president of the Weymann American Body Corporation. Mr. Ruehlmann concluded his address with motion pictures of the process of manufacturing all-steel bodies.

The discussers conceded advantages to the all-steel body and that it has an important place in mass car-production but many favorable comments were made on the fabric body and prominent men in the industry predicted that it will come into extensive use in this Country as it has in Europe. One of these was Thomas J. Lítle, Jr., president of the Society, who was recently appointed chief engineer of the Marmon Motor Car Co. and is now a resident of Indianapolis. He was given a hearty welcome on behalf of the meeting by Ralph Teetor, chairman of the Section, who presided, and made a brief speech at the opening of the session in which he congratulated the Section upon the spirit and interest shown at the local meetings, which is due, he said, to the selection of interesting subjects and speakers.

STEEL CONSTRUCTION STRONGEST AND SAFEST

Mr. Ruehlmann described the construction of the all-steel bodies, which was set forth in detail with illustrations in a paper[1] by Edward G. Budd and J. Ledwinka which was presented at the 1925 Annual Meeting and contrasted them with the composite wood and steel body, declaring them stronger, lighter, quieter and safer, just as the all-steel railroad car possesses these advantages over the wooden coach. He gave electric and gas welding some of the credit for this superiority, as it adds to the strength of the metal joints instead of decreasing it, as the mortising, screwing and bolting of wood does. He pointed out that hollow steel frame-members of much smaller cross-section have equal or greater strength than wood members and afford more unobstructed vision for the driver and more interior space in a body of the same external dimensions.

Production advantages mentioned are reduction in production time, in storage space for material and in fire risk, and the speed at which the bodies can be sent through the drying ovens after enameling or lacquering. After years of experimenting, he said, a steef roof of the same weight as a wood roof and which produces no drumming noise has been developed. A different way of upholstering steel bodies had to be devised to eliminate the use of tacks, and thus a new method

[1] See THE JOURNAL, February, 1925, p. 219.

of trimming was developed that can be done by unskilled men and reduces the cost. The cost of dies for producing steel bodies is heavy but when the run on a certain model is large the cost of the dies per body is brought down to a few dollars. The steel body will outlast any chassis on which it is mounted, asserted Mr. Ruehlmann, and the joints never loosen and become noisy. Within 2 hr. after the sub-units of the body are placed in the final-assembly jig the complete body, minus upholstery or trimming, is ready to be sent to the paint shop.

Flexible Bodies Are Light and Noiseless

Mr. Steinbrugge told how the Weymann flexible fabric bodies are built in a series of parallelograms, with a framework of wood joined together by plates of cold-rolled steel which give a certain flexibility. Wood in contact with metal is noiseless, he said, and rubber pads can be placed between the base of the parallelograms and the walls of the chassis frame to which they are bolted. Floor-boards and seats are bolted directly to the chassis frame. Doors are built with flexible joints and have flexible hinges. Imitation leather is used as the covering for the body frame. Details of the construction were given in a paper[2] by George W. Kerr, which also was presented at the 1925 Annual Meeting.

Among the manufacturing advantages mentioned by Mr. Steinbrugge are: (*a*) the complete unassembled body can be stored or shipped in small space, (*b*) the covering is cut in lots of 25 to 50 at a time by electric cutting-machines, (*c*) the chain method of assembling is easily adopted, (*d*) no painting is necessary, and (*e*) models can be changed at slight expense merely by making a set of blueprints, rejigging in the wood-working shop, providing joints of a few new shapes and some new pasteboard patterns in the leather-cutting department. He listed about 40 European car-builders who now use flexible bodies.

Light Weight Will Improve Car Performance

Spirited discussion followed the presentation of the papers. Mr. Lítle predicted rather general use of the fabric body in this Country within a few years because the fabric roof and sides will not act as resonators for the vibration of the powerplant, hence the body will be quiet, and because the light weight will make possible better car performance without increasing the powerplant. James A. Daugherty, of the Robbins Body Corporation, said,

> If we can save 400 or 500 lb. in the weight of a body, it means a great deal to the chassis manufacturer as well as to the man who buys and operates the car. There always will be a demand for bodies that must be designed and built in quantities ranging from 45 to 500 and 5000. It is impossible to build these in the all-steel construction and it seems to me that this is the field for the fabric type.

Admitting that the steel body is ideal for large production, Mr. Roos, of the Studebaker Corporation of America, said it is a serious engineering problem to lay out a chassis hoping to get a certain performance and to have this performance handicapped by excessive body-weight. From what he has seen he thinks the fabric body will add to the performance of cars and probably will take a legitimate place in the industry. Fred S. Duesenberg said that he looks for wonderful results from the fabric body if it is built right, as, due to the construction and light weight, it should give easy riding and make the car easy to handle on the road.

Benjamin Twyman, now connected with the Weymann American Body Corporation, stated that the construction lends itself to any of the designs of present bodies in either composite or all-steel construction. All of the weight saving is effected above the deck line, which brings the weight of the car mass lower and makes it safer. The body weight can be reduced from 40 to 60 per cent, which means that lighter chassis springs can be used, that tires can be smaller or will give more mileage, and that the car can run more miles per gallon of gasoline; also that the car will be more comfortable and quieter. The tool cost for this type is very small and style changes can be made easily. The cost of recovering the Weymann-type body is little more than the cost of repainting other bodies.

Steel Body Weights Have Been Reduced

Mr. Ruehlmann gave the weights of three fully trimmed all-steel bodies as: coupe, 540 lb.; coach, about 740 lb., and sedan, 800 lb., and said that the weight of the Dodge sedan 3 years ago was 1000 lb., which shows a reduction of 200 lb.

Fred E. Moskovics, president of the Stutz Motor Car Co., told of going to Europe last spring to investigate the Weymann body and being so impressed that he ordered a Stutz chassis sent over to be fitted with one, which he has used for 4 months. This car, he said, lapped the Indianapolis Motor Speedway 2-mile track in 1 min. 56 sec. the day before the meeting and another car of the same model fitted with a composite body took 2 min. 8 sec., or 12 sec. more. The lowering of the center of gravity by keeping the top light had decreased the chance of overturning the car by 80 per cent, he believes. He referred to the hoops on the Fords used for playing automobile polo, which frequently roll over without injuring the driver, and said that any desired strength could be built into the frame of the fabric body but he would rather eliminate the chance of rolling over in a car. Bolting the seats to the car frame contributes to the flexibility of the body, and he compared riding in it to riding on a chassis with the seats bolted to the frame and a light cover thrown over the car.

[2] See The Journal, February, 1925, p. 215.

ELY URGES REDUCTION OF VARIETIES

Cleveland Section Told How Simplification Paves Way for Standardization

When a standardization body has created its standards the job is not by any means finished; the work is just begun, because standards are not worth much unless the people use them, E. W. Ely, assistant chief of the division of simplified practice of the Department of Commerce, told members of the Cleveland Section in an explanation of and a plea for simplification at the Section's monthly meeting on Dec. 20.

The meeting was held at the Hotel Cleveland following a members' dinner and an unusual program of entertainment. The technical session was opened by John H. Jaschka, vice-chairman of the Section, who reported that Chairman T. V. Buckwalter was far from well and probably would have to go South for some weeks to regain his usual good health. In accordance with a new practice of the Section instituted last autumn, a member of the Governing Committee, Eugene Bouton, was called to the chair to take charge of the meeting. Mr. Bouton introduced Mr. Ely as the speaker of the evening.

Enormous Avoidable Waste in Industry

Immediately after the World War everybody was complaining of the high cost of living and attributing it to scarcity of raw materials and to profiteering, said Mr. Ely, but Herbert Hoover, Secretary of Commerce and president of the Federated Engineering Societies, thought the spread between cost of production and cost to the consumer might be due to another cause. Therefore a committee of six was organized to make a survey of six typical industries. Based on the total value of goods produced in the year 1921, they estimated that the avoidable waste in industry amounted to about $30,000,000,000 per year.

Reduction of varieties and sizes was insisted upon during the war by the War Industries Board, said Mr. Ely, who showed many lantern slides of charts to illustrate the great reduction effected in a number of items. The automotive industry has practised standardization to a certain degree, he said, and it is due perhaps to that practice that prices have gone down. The General Motors Corporation, for example, has reduced the bolts and nuts it uses from 13,000 to 2100 sizes.

Drawing a distinction between the two, Mr. Ely said that

standardization seeks to annotate the sizes and dimensions and the performance of products, whereas simplification seeks to eliminate sizes and varieties that are superfluous and concentrate on those that are in greatest demand. Simplification decreases inventories, idle investment, seasonal fluctuations, production costs, and selling expense; it increases sales, turnover, working capital, employment stability, continuity of operation, quality of product, profit to manufacturer, distributor and consumer; and conserves manpower.

Waste Elimination Increases Purchasing Power

Simplification prepares the way for standardization; each can help the other, declared Mr. Ely. Sixteen reasons account for the vast waste in distribution. The Division of Simplified Practice is working on No. 5, which is "too many varieties"; other groups are working or will work on all of the other 15. Management has the greatest opportunity and responsibility for eliminating waste. Wages have gone up faster than the cost of living since 1914, and this spread represents purchasing power. The manufacturer, by practising simplification, puts himself in position to obtain more of these dollars.

A re-survey of half a dozen simplification projects after 1 year showed an average of 79 per cent adherence to the simplified lists, which is good. Can you say as much for S. A. E. Standards? asked Mr. Ely. To answer this question a conference of manufacturers, distributors and users of spark-plugs, pistons, piston-rings, brake linings, and roller bearings will be held in Detroit in February. The speaker said it is the responsibility of engineers, draftsmen, managers, future executives, and salesmen to make the most of what they find at hand and not create something new and that their chief value to their employers lies in this direction.

Considerable discussion followed presentation of the address and was participated in by Chairman Bouton, Mr. Ely; Mr. Loew, of the Loew Mfg. Co.; J. W. Saffold; B. H. Blair; E. W. Weaver; C. M. Taylor, and Ferdinand Jehle. It pertained largely to waste in advertising, the height of bumpers, standardization of frequency of electric current, and the possibilities of securing cooperation of the consumer in a program of simplification.

McCOOK FIELD INSPECTION TRIP

Dayton Section Members and Dayton Engineers Inspect Shops and Laboratories

An inspection of the equipment of the material branch of the Engineering Division of the Army Air Corps at McCook Field, Dayton, was made on Dec. 10 by more than 100 members and guests of the Dayton Section and of the Dayton Engineers' Club. Following the inspection, a dinner was served at the Post Restaurant, Brigadier-Gen. William E. Gillmore and other executive officers of the Materiel Branch being guests of honor. Music was furnished by a five-piece orchestra.

Small groups of guests were conducted through the various hangars, shops and laboratories during the inspection in the afternoon. A special demonstration of recent aircraftt engines was staged in the powerplant laboratory. A Packard 2A-2500, a geared Packard 2A-1500, a Curtiss D-12, and an air-cooled inverted Liberty engine were set up on dynamometer stands ready for running. As each group of guests passed through the laboratory, a description was given of the different engines. A demonstration was also made of the full-throttle performance of the Packard 2A-2500 and the Curtiss D-12 engines. The various new types of airplane under development by the Air Corps were inspected and described during the trip through the hangars.

The guests assembled in the auditorium in the late afternoon and were entertained by viewing a special motion-picture, showing the activities of the Air Corps, that was filmed under the direction of General Gillmore. Another picture entitled The Airdales, with Capt. L. G. Meister and Lieut. W. H. Brookley as star performers, furnished considerable amusement. F. G. Shoemaker, of the Powerplant Branch, summarized recent development in aircraft engines as exemplified by the various engines exhibited.

In his address following the dinner, General Gillmore outlined the organization of the Materiel Branch. He emphasized the importance to the Government and to the automotive industry of having the executive offices of the Materiel Branch concentrated in a single unit.

RECENT DIESEL-ENGINE DEVELOPMENT

Late Types of Stationary and Marine Diesels Described to Milwaukee Section

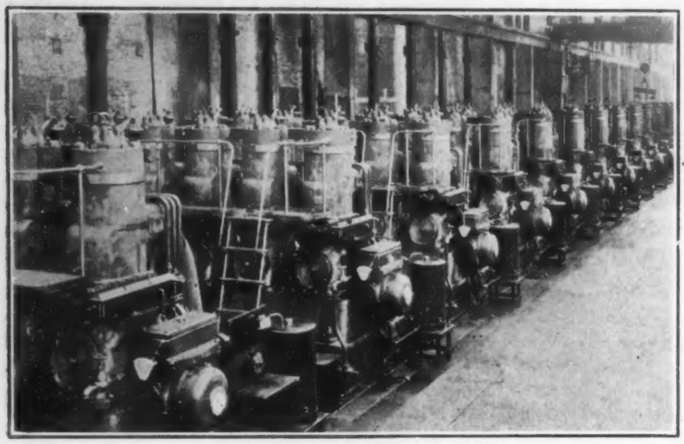
F. P. Grutzner

Comprehensive Diesel-engine service in the stationary-engine and the marine-engine fields was featured at the Milwaukee Section meeting that was held Dec. 1. F. P. Grutzner, engineer in charge of installation for Fairbanks, Morse & Co., Chicago, presented a paper replete with illustrations of the Diesel-engines built by the company he represents and described the several types of engine it produces.

Mr. Grutzner reviewed briefly Diesel-engine history and stated that his company has developed in the last 2½ years a line of Diesel-engines not only suitable for different kinds of stationary installation but other types specially designed for direct propeller-drive on ships, for double-end drives on ferry boats, for stern-end drives on river boats, and for use on dredges. All the engines are of the single-acting two-cycle type. Up to 360 hp., the crankcase is used for scavenging air and above 360 hp. the engines are supplied with a special scavenging-pump. The stationary engines range from 40 to 720 hp. and those of the marine

Tug Equipped with Two 360-Hp. Diesel Engines

Arranged Side-By-Side and Installed on the Tug Mahoe, in the Harbor at San Francisco, the Engines Made a Non-Stop Run to Honolulu after a Short Test-Run of Only 8 Hr. Engines Are Shipped from the Factory Completely Assembled Except That Some Exterior Piping Is Removed



TESTING STATIONARY DIESEL-ENGINES
Completed Engines Are Tested for 1 Week. The Long Test Is Made So That the Customer Receives the Engine Already "Run-In" and Ready To Operate at Full Load

type from 60 to 720 hp. The marine engines are all heavy-duty types, the weight being approximately 160 lb. per hp.

OPERATION

Air is sucked into the engine through a series of ports arranged on the side of the frame and goes through an automatic suction-valve into the crankcase where it is compressed to about 1¾ lb. per sq. in. and forced through the ports as they are opened by the piston. A device on each piston forces the air into the cylinder, scavenging the cylinder and making it ready for the next cycle. Then, after the ports are covered, the charge in the cylinder is compressed to about 500 lb. per sq. in. Shortly before dead-center, the fuel is injected into the combustion-chamber and the pressure rises from around 50 to 60 lb. per sq. in. to be about 150 lb. per sq. in. All the fuel is burned near the dead-center position, and the expansion is similar to that in a gasoline or a steam engine until the piston uncovers the exhaust-port again for the pre-exhaust of the charge, later uncovering the intake-port, and then the cycle is completed.

Two-stage combustion is used. The fuel is checked in the first chamber and the charge is burned to carbon monoxide. Then the gas is forced through a neck in the main cylinder as soon as the piston starts to move down, and the gas is burned to carbon dioxide. This change of the fuel into carbon monoxide and then into carbon dioxide has not been proved scientifically but it is believed to be true. The upper combustion-chamber is thoroughly water-cooled although, even if it were not, the temperature would not become very high. But the temperature in the second combustion-chamber is very high, reaching somewhere around 2500 deg. fahr. For that reason, the second combustion-chamber is a special casting, water-cooled throughout.

Detailed description of the engine parts and of special features relating to design followed. An interesting feature mentioned was that of the development of a stern-wheel drive for river boats. Stern wheels must turn slowly, but Diesel engines run at high speed. Therefore, the engine for such service is installed far forward in the boat and a long shaft supplied with flexible couplings is run aft, the speed-reduction between the shaft and the stern wheel being made through bevel gears.

FUTURE DEVELOPMENT

A tendency toward higher rotative as well as higher piston-speeds is noted by Mr. Grutzner, who said that piston speeds so far are very conservative. He believes that great increase in piston speed can be made without endangering the reliability or the safety of the engine. He mentioned also the necessity of increasing rotative speed to reduce first cost, weight and over-all dimensions. He predicted that a small-size Diesel-engine having revolutionary attributes as to speed and weight would shortly be announced. He said that there seems to be no limit to the size and power of large Diesel-engines, citing as an instance the 15,000-hp. nine-cylinder double-acting two-cycle type Diesel-engine now running in Hamburg, Germany, which is 38.7 ft. high.

Questions asked and answered during the discussion related mainly to some of the fine points of design and to further elaboration of the descriptions already given.

MODERN METHODS OF CLEANING

Body Washing and Steam Cleaning of Chassis Told to New England Section

Varnished bodies should never be washed with anything but cold water, and steam cleaning is one of the most effective and economical methods of cleaning chassis and chassis parts, members of the New England Section were told by Guy Gregory, mechanical engineer of Oakite Products, Inc., at its monthly meeting on the evening of Dec. 21. The meeting was held at the Engineers' Club in Boston and was preceded by a members' dinner.

Following the calling to order of the meeting by Chairman Glenn S. Whitham, a nominating committee consisting of five Past-Chairmen of the Section was elected unanimously, as follows: H. E. Morton, J. A. Moyer, V. A. Nielsen, R. E. Northway, and M. R. Wolfard. Chairman Whitham was elected to represent the New England Section on the Society's Sections Committee during 1927.

Mr. Gregory's paper on Cleaning Automotive Equipment dealt with the value of keeping motor-vehicles clean and attractive in appearance, the technical problems of cleaning and methods, materials and equipment used for the purpose.

Cleaning methods were divided by him into four general classifications according to their action, as (*a*) by mechanical abrasion, (*b*) by mechanical solvents, (*c*) by chemical solvents, and (*d*) by emulsification. The last is carried out by the use of a solution of the emulsifying agent in hot water and is practically displacing the use of gasoline and oil solvents, which involve considerable fire hazard and leave an undesirable surface for subsequent operations.

Clear, cold running water and a sponge alone should be used, however, on varnished surfaces, followed by drying with an air jet. Oil spots and stains should be removed with a sponge moistened with a solution of Composition No. 1 prepared by the company he represents, he said and then air dried.

Running-gears and their component parts should be cleaned with steam and the same compound directed against the parts by a steam gun at a pressure of not more than 40 or 50 lb., care being taken not to hold the nozzle too close to the painted parts, as the temperature of the spray ranges from 130 to 140 deg. at a distance of 10 or 12 in. A complete phaeton chassis can be cleaned in from 20 to 45 min. by this method and an engine in 15 to 25 min. A truck chassis usually requires from 1 to 3 hr., depending upon its condition. The same method saves considerable time in cleaning disassembled parts but is somewhat more expensive than dipping in a cleaning tank.

Methods of cleaning radiators and removing old paint were also described, and Mr. Gregory concluded with a plea for improvement of the working conditions of the washers.

AUTOMOBILE LAUNDRY WASHING-SYSTEM

The discussion was opened by Mr. Hyland, of the Motor Mart in Boston, who said that he found that compressed air does not give pressure enough to wash the accumulated mud and grit from a body and that an hydraulic pump does the work quicker. Air does a better drying job after washing than does chamois, as a man with an air-gun is in no danger of scratching the finish and can blow the water from around doors and window sashes.

His company, he said, is testing the automobile laundry system of washing. The cars are run on long tracks, each of which holds four cars, and are first cleaned in pairs with a vacuum cleaner. Next they are sprayed with kerosene and air to soften the grease, then water is applied with pressure

from an hydraulic washing-machine. The succeeding operation is spraying with a gun and washing with a sponge, soap and water. The cars are then dried with compressed air and the glasses cleaned and finally are run into a closed room where the body and nickeled parts are polished. By this method 12 men can clean 200 cars in 10 hr.

Answering Mr. Whitham, Mr. Hyland said that the kerosene spray is used only on the chassis, but on some cars a mixture of kerosene and water is used on the body after it is cleaned, and then after going over it with a sponge and cold water and rinsing and drying, a good finish results. A little time-saving trick which is used on cars that are washed regularly is to rub the under sides of the fenders with a sponge soaked in a strong solution of kerosene and water. This leaves a film of oil to which dust and mud thrown up by the wheels adhere. When the car is next brought in for cleaning, knocking on the fenders causes the dirt to drop off, after which both cold and warm water at about 100 deg. is used under 300-lb. pressure from the pump.

Kerosene under air pressure from a Hansen motor gun has been found very satisfactory for cleaning engines, he said. The price received for washing a car is $2.50. A car is washed from start to finish, including polishing the nickel, in about 20 min. Water stains on the nickel cannot be removed, said Mr. Hyland, without taking off nickel, too.

Kerosene Affects Brake-Linings

W. M. Clark asked if trouble had not occurred from kerosene affecting brake-linings, to which Mr. Hyland replied that he had had many complaints that the kerosene hardens the brake-linings so that the brakes do not hold but that other customers reported that their brakes held too much. Mr. Clark suggested as a safeguard that the brake-bands be set tight before washing the wheels to prevent the kerosene getting on the linings.

K. T. Brown said that in the Packard service station the chassis are cleaned with a washing machine giving a pressure of 300 lb. and using an Oakite solution. The same method is used for cleaning engines, but the ignition system is covered. Whereas a man formerly spent 5 or 6 hr. cleaning a chassis with a knife and a brush, one man with the machine does the work in about ½ hr. and no damage results. Mr. Brown declared that he intends to heat the solution in the tanks with live steam and spray it under pressure from a pump. With washing machines some trouble arose from the solution injuring the valves. By use of the heated tanks and pump, and with the cars on a washing stand, the hood will be removed and the engine washed before any mechanical work is done. He estimates that the whole engine can be cleaned in 10 min. The solution is not strong enough to injure the paint on the chassis.

A simple expedient employed in the Charles Street Garage for cleaning a dirty engine, said Mr. Whitham, is to sop the top of the engine with kerosene and let it stand for 20 min., then turn a stream of cold water under pressure on it, keeping the engine running and covering up nothing. The engine is thus warm enough to dry off quickly.

In response to various questions by members, Mr. Gregory said that for removing paint a solution of 6 or 8 oz. of Oakite Test X is strong enough to remove the paint as fast as the solution is flowed on, after which it is rinsed off with plain water either warm or cold; a special preparation is made for cleaning aluminum without tarnishing and this can be used on other metals also; the strength of solution in dipping tanks is maintained almost indefinitely, as the Oakite does not evaporate, neither does it break up oil as alkali does; no provision is made for filtering air used for drying, as a strainer is used on the air compressor; and Oakite is odorless.

Mr. Wolfard suggested that the air be heated to prevent condensation of moisture. Mr. Whitham exhibited an air gun that is used in the Middle West, sells for $50, and is very efficient, he said, as it has a high velocity and does better work than straight water-pressure.

MOTORCOACH TRANSPORT ANALYZED

Washington Section Studies "Riding-Habit" and Motorcoach-Operation Data

Interstate motorcoach-transportation, statistical data on the "riding-habit" of the public and the influence of motorcoach troubles on operation were the meaty subjects discussed at the motorcoach transportation dinner and meeting of the Washington Section that were held on Dec. 15 at the Hotel Hamilton, City of Washington. Lieut-Col. J. Franklin Bell, engineer commissioner of the City, was the guest of honor. In the course of his address he advocated a central motorcoach terminal for cities as a junction point for all interstate motorcoaches, where cooperative information regarding time schedules, fares, routes, frequency of trips, and the like would be available and where travelers could transfer from one route to another. In his opinion, selling the idea of motorcoach transportation to the public has not yet progressed as far as it should.

Two papers were presented at the technical session; one was by E. D. Merrill, president of the Washington Rapid Transit Co., who analyzed statistics relating to the so-called riding-habit of the people in various cities; and the other was by E. Pardoe, of the Central Traction Co., who presented data on motorcoach operation derived from comprehensive investigations.

Riding-Habits of the Public

Mr. Merrill said that the American public demands time-saving and comfort as prime factors of transportation. His statement was based upon an analysis of "riding-habit," determined in numerous cities by dividing the number of revenue passengers per year by the population. Southern cities have a higher riding-habit than Northern cities. Rate of fare, business conditions as indicated by increases and decreases in bank clearings, climate, and nearness of the service to the homes have an important effect on riding-habit according to extensive tests. The evidence of the people who

SCHEDULE OF SECTIONS MEETINGS

JANUARY

10—Metropolitan Section—Automobile Show Dinner Meeting—What's New at the Show by the chief engineers of the exhibiting companies

14—Southern California Section—Trailers and Semi-Trailers—C. H. Mason, G. L. Knox and other trailer building companies' representatives

18—New England Section—Gas-Electric Motorcoaches—M. E. Toepel

20—Dayton Section—Some Thoughts on the Use of Our Highways—Dr. H. C. Dickinson

actually ride is considered conclusive as to when and where the majority wishes to ride.

As to how much territory is required to produce enough traffic to justify a motorcoach or a street-car line, Mr. Merrill stated that the distance, in blocks, between lines of transportation is determined by dividing the frequency of trips times length of trip times cost per mile by rate of fare times riding-habit times density of population. Since rate of fare and riding-habit are denominators it is clear that, the other factors being constant, the distance between the needed lines is decreased and the transportation is brought nearer to the door of the home, which increases riding-habit automatically. Because it is uneconomical to install tracks on adjacent or too nearly adjacent streets, motorcoach service for the outlying territory of cities is indicated. Mr. Merrill regards motorcoaches and street-cars as co-workers rather than as competitors.

Motorcoach Statistics

Mr. Pardoe presented statistics, stated costs per mile of operating, suggested opportunities for cost reductions, and listed troubles causing delays en route for 10 months of 1926 motorcoach operation in the City of Washington. His data include the facts that approximately 6500 companies now operate 25,000 common-carrier motorcoaches over 250,000 miles of route in the United States. Of these, 339 electric-railway companies operated 6500 motorcoaches over 15,000 miles of route; that is, 26 per cent of the total number of motorcoaches and 6 per cent of the route miles. Thirty-one steam railroads operate 350 motorcoaches over 4500 miles of route. In 1921, only 16 electric-railway companies were operating a total of 73 motorcoaches.

Motorcoach operations class themselves naturally into three main groups, said Mr. Pardoe; city service, interurban service and as feeders to the electric railways and the steam railroads. Generally speaking, the first two classes are profitable, but numerous lines in these classes have failed or are losing money because they were started without proper study and surveys or because of franchise reasons. Mr. Pardoe said also that the feeder lines are nearly always heavy losers. They are operated for the convenience of the public because the loss from operating motorcoaches is not as great as would be true of rail operations, particularly where heavy expenditures need to be made for the extension of tracks. Since the majority of motorcoach operations by electric railways is in the feeder class, the vital necessity of reducing operating cost is evident.

TABLE 1—OPERATING COST PER MILE FOR A 29-PASSENGER SINGLE-DECK MOTORCOACH

	Cents per Mile
Maintenance	
Bodies	0.50
Chassis	1.40
Tires and Tubes	1.75
Depreciation	5.00
Buildings and Miscellaneous	1.00
	9.65
Garage Operation	
Employes, Supplies and Expenses	2.25
Transportation	
Superintendence	0.90
Chauffeur's Wages	6.00
Fuel	3.35
Lubricants	0.25
Miscellaneous	0.50
	11.00
Traffic Promotion and Advertising	0.10
General	
Officers' Salaries	0.60
Injuries and Damages	0.15
Insurance	0.05
Miscellaneous	0.20
	1.00
Taxes	1.00
Total Cost per Mile	25.00

Table 1 giving data on the operating cost per mile for a 29-passenger single-deck motorcoach was presented by Mr. Pardoe as a fair average that will serve to indicate how automotive engineers can reduce operating costs.

An analysis of motorcoach troubles that caused delay to the service was also made by Mr. Pardoe for the first 10 months of 1926. The analysis is presented as Table 2, it being stated that 95 per cent of the item "collisions" comprises minor cases of scraping fenders and of running into rear bumpers.

TABLE 2—MOTORCOACH DELAYS DUE TO TROUBLES EN ROUTE FOR 818,186 MILES

	Total Number	Number per 10,000 Miles
Axles	13	0.18
Battery	7	0.08
Brakes	6	0.07
Carbureter	6	0.07
Clutch	10	0.12
Collisions	107	1.31
Differential	6	0.07
Gasoline Line	11	0.13
Generator	4	0.05
Gear-Shift	9	0.11
Ignition	40	0.50
Lights	6	0.07
Springs	9	0.11
Steering Apparatus	4	0.05
Tires	45	0.55
Transmission	4	0.05
Vacuum Tank	11	0.13
Miscellaneous Mechanical	20	0.25
Miscellaneous	60	0.73
Total	378	4.61

AIRCRAFT-POWERPLANT DEVELOPMENT

Prevailing and Projected Types Analyzed Comparatively for Buffalo Section

Recent developments in aircraft powerplants were described by Lieut. C. C. Champion, Jr., U. S. N., of the engine section, Bureau of Aeronautics, Navy Department, at the meeting of the Buffalo Section that was held on Dec. 7. He said in part that during the World War naval aviation confined its activities to training and to coastal patrol. This limited operation was necessitated by the small amount of material suitable for operation over water, the strategical and geographical situation which determined the nature of the naval operations, the very limited performance of seaplanes of that period, and the fact that warships were not equipped for handling aircraft or prepared for aircraft co-operation. The great naval value of aircraft had made itself apparent by the end of the War, and naval aviation was then made part and parcel of the fleet. The lines along which materiel is to be developed must be in conformity with the service which the materiel will be called upon to render.

Fighting airplanes are required to gain and maintain control of the air. Observation airplanes are used for short-range scouting and also for controlling long-range fire of capital ships by reporting the fall of shot to the ship by radio. For torpedo and bombing work, the first requirement is large weight-carrying capacity. This dictates the use of an airplane of considerable size which, in practice, is found to be about the maximum size that can be handled conveniently aboard ship. A definite need exists for scouting and patrol operations at great distances from the fleet or base, requiring flights of great range and duration. In the present state of development, the multiple-engine flying-boat seems best suited for this purpose. Training airplanes need have only limited performance and must require some skill of the pilot. They should have low first and maintenance costs but weight, within reasonable limits, is not important. The difficulties of providing for operation on short notice, irrespective of the state of the sea, as well as

the limitations on aircraft imposed by shipboard operation, exert a profound influence on the development of materiel. The problem divides itself into the handling aboard ship, the take-off, the landing and the hoisting-in, and the speaker went on to enumerate the difficulties incident to their satisfactory accomplishment.

Comparison of Engine Types

Regarding aircraft engines for naval use, the two fundamental differences are the arrangement of the engine and the method of cooling. The two accepted arrangements at present are the radial and the in-line, and the cooling is by air or by water. An analysis of troubles experienced with water-cooled engines shows at once that a large proportion of the failures which have occurred are failures of some part of the water system. Besides being the source of frequent failures, the water-cooled system adds at least ½ lb. per hp. to the installed weight of the powerplant. The weight per horsepower, dry, of the most modern types of engine, is approximately the same, but it is a misleading comparison in that the chief concern is the weight of the powerplant when ready to fly. Air-cooling eliminates the troubles and the additional weight due to water-cooling and, in the matter of mechanical dependability, there seems to be little choice between the engines proper of the two types. The elimination of the water-cooling system results in a net gain of from 25 to 40 per cent in weight. From the viewpoint of fuel economy, the two types are equally good.

Three distinct types of engine are now being produced for the Navy. Two models, practically identical in design and construction, are Packard engines of from 500 to 600 hp, and of 770 hp. respectively. Each is a 12-cylinder 60-deg-V type. The Wright nine-cylinder Model J-5 radial air-cooled engine is the latest development for the Navy in the 200-hp. class. The Pratt & Whitney Wasp engine is the latest development in American aircraft engines to be put into production. Lieutenant Champion said that he flight-tested the No. 2 Wasp engine at an engine-speed of almost 2500 r.p.m., and that he flew a 120-mile race with such an engine; the engine-speed was 2270 to 2280 r.p.m.

After detailing various phases of flight experience, the speaker discussed superchargers, starters, ignition, the fire hazard of airplanes, gearing of airplane engines, heavy-oil solid-injection compression-ignition engines for aircraft use, and steam powerplants for the same purpose. He said that the solid-injection engine lends itself most favorably to the propulsion of rigid airships and that the steam powerplant is not worthy of serious consideration, for aviation purposes in any form which has been devised to date. In conclusion, the speaker stated that, as funds became available for development and for the purchase of new equipment, the results of the Navy's policy have been to produce a condition of stability in an industry which, without such support, would have been reduced to a chaotic state; a steady progress in engine design and engine building that has resulted in a line of excellent powerplants produced to meet naval needs; one engine to date which has found much favor with commercial users; and the development of a line of engines that should fill commercial as well as naval needs.

PENNSYLVANIA TOLD OF DEVELOPMENTS

Section Hears about New Truck, Six-Wheel Suspension, Fuel and Other Things

What We Have Done This Year was the topic on which representatives of seven companies talked at the monthly meeting of the Pennsylvania Section on the evening of Dec. 14. The meeting was held at Kugler's Restaurant and followed a members' dinner. Norman G. Shidle presided as chairman of the meeting. Secretary A. Gelpke, of the Section, reports that A. L. Clayden, of the Sun Oil Co., told about a special lubricant developed by this company during the year and which is used successfully for eliminating noise from steering-gears. He also spoke of the recent development by the company of an antiknock fuel. In a test conducted for 1 month by a taxicab company with a fleet of 25 cars using the antiknock fuel and 25 using straight commercial gasoline, the speaker said that the fleet using the new antiknock fuel showed a fuel saving of 13 per cent.

B. B. Bachman, of the Autocar Co., told about the new 1½-ton truck developed recently by this company. New features embodied in the six-wheel suspension for motorcoaches and motor-trucks by the Six-Wheel Co. were explained by E. W. Templin. The construction and application of some new SKF roller bearings designed and produced by the Hess-Bright Bearing Co. especially for heavy loads were described by C. L. Drake, of that organization. S. W. Andrew, of the Mitchell Specialty Co., told about a new electric car-lock recently produced by his company.

The new eight-cylinder engine developed by the Lycoming Motors Corporation was described by E. D. Herrick, who said that it has a bore of 2¾ in., a stroke of 4¾ in., compression of 4.9 lb. per sq. in., and develops 62 hp.

Apparatus of different kinds built by his company for testing automotive parts was described by Thorsten Y. Olsen, of the Tinius Olsen Testing Machine Co.

LATEST DEVELOPMENTS IN AERONAUTICS

Description of Modern Airplanes Thrills Southern California Section

The Latest Developments in Aeronautics entertained the members of the Southern California Section at its regular monthly meeting at the City Club, Los Angeles, on Dec. 10. Mr. Kindelberger, chief engineer of the Douglas Co., Santa


N. G. Shidle


A. L. Clayden


E. W. Templin


B. B. Bachman

The Chairman (at the Extreme Left) and Three of the Speakers at the December Meeting of the Pennsylvania Section

Monica, Cal., builder of airplanes, spoke on the Development of Metal Struts for Airplanes; W. B. Kinner, president of the Kinner Airplane & Motor Co., Glendale, Cal., explained the Commercial Use of Sport Planes, taking as an illustration some of the machines produced by his company; and Lieut-Com. L. B. Richardson, Construction Corps, U. S. N., Clover Field, San Diego, Cal., described the requirements of a modern airplane in order that it can meet the demands made upon it, the types of airplane at present in use in the United States Navy, the details of their equipment, and the experiments that are being made with a view to meeting the demands of the future. Eugene Power, chairman of the Section, presided and the discussion was led by Ethelbert Favary.

Requirements of Modern Airplanes

The demands made upon airplanes of all classes are continually becoming more exacting, said Commander Richardson. They must carry more and heavier bombs, machine guns, ammunition and radio sets; must go faster, climb higher and remain in the air longer; must be designed for minimum structural weight yet sturdy enough to withstand landings and take-offs in heavy seas and to ride out storms at buoys; must be braced against the high acceleration of catapulting and the deceleration of deck landings; must be easily maneuverable; have a minute turning-circle, a quick take-off and slow landing-speed; must go into a tail-spin easily and come out of its own accord; must have sufficient stability to fly "hands off" yet must answer its controls with lightning speed and precision; must have ample space for the comfort of each member of the crew and unobstructed vision in all directions; must protect the personnel against injury in case of possible crashes; and must have reinforcements and braces so that heavy-footed mechanics can clamber about it without damaging it unduly. Bombing airplanes must have all the instruments for discharging bombs arranged in orderly fashion and readily accessible behind a glass window fitted with a windshield-wiper; must carry parachute flares, landing-lights, smoke bombs, baggage of the crew, emergency rations, anchor and line, engine tools, and spare parts; must have folding wings to allow ready stowage aboard ship; and must allow rigging for flight in 3 min. When rigged the total wing-spread must not exceed the dimensions of the elevators used to hoist the airplane to the flying-deck of the carrier. The hull of floats of flying-boats must not soak up water, when anchored for an entire season, yet, if made of metal, must not corrode.

The powerplants, including not only the engine and propeller but also the fuel, oil and water systems with all tanks, pumps, piping, coolers, and heaters, continued Commander Richardson must fulfil many requirements also. The engine must weigh not more than 1¾ lb. per hp. yet must be rugged enough to run 300 hr. between overhauls without breakdown, though running a large part of the time at nearly full power. The fuel, oil and water pumps must be light, compact and reliable. The propeller must be thin enough to be light and efficient, yet must withstand the attacks of spray and rain. The fuel and oil tanks must not leak under constant vibration, must be made of a material not affected by benzol or ethyl fluid, yet they cannot be of copper or tinned iron, because these materials are too heavy. A device must be provided for heating the oil quickly when starting, yet must keep the oil cool during flight. The radiator must be of the lightest design yet leakproof, must be located where it will produce the best cooling yet not interfere with the pilot's vision or cause undue air-resistance or be a menace to the pilot in a crash.

Commander Richardson then explained what had been done with a view to meeting these requirements and the compromises that had been necessary. Progress has been made, he said, along three principal lines: aerodynamics, structure and suitability for the purpose intended. Advances in aerodynamics have consisted mainly in refinements of details and in ingenious rearrangements of wings, control surfaces and their bracing. In structural progress, the trend has been toward the substitution of metals, such as duralumin, for wood.

In adaptation to specific purposes, Commander Richardson explained that the five classes of airplane being developed for the Navy comprised airplanes for training; for fighting or pursuit; for gunnery observation or short-distance scouting; three-purpose airplanes for torpedo-dropping, bombing and long-distance scouting; and patrol airplanes. Development of the engine and of the powerplant accessories has resulted in decreasing the weight of the engine; in increasing its power, reliability and ruggedness; and in the use of superchargers and metal propellers. Experiments now being made in aeronautics are along the line of reducing the take-off and landing-speeds, of reducing the drag in seaplanes by designing floats offering less air-resistance, of operating the control levers by motors in much the same manner as the steering-gear of ships is now operated, of developing some process for preventing the corrosion of duralumin, of using adjustable-pitch propellers on supercharged engines, and of reducing the excessive air-resistance and weight of large water-cooled engines by a method of steam-cooling.

The Lines of Future Development

Looking into the future, Commander Richardson said that the tendency is to abandon the float type, or seaplane, in favor of the hull type, or flying-boat, because of the greater seaworthiness of the latter. The three-purpose airplane will probably be superseded by three distinct types as the numbers of aircraft in the Navy increases, in other words, the scouting will be done by amphibians of long range, the bombing by land airplanes of good climbing ability, and the torpedo launching by flying boats of high speed.

The commercial value of sport airplanes was described by Mr. Kinner, who called attention to the fact that, as such planes are designed to carry one or two passengers in addition to the pilot, their carrying capacity would be from 165 to 330 lb. Although their horsepower is not high, they might be of service in time of war in carrying messages, food or ammunition behind the lines. In time of peace their value would lie in transporting traveling salesmen, doctors, ranchers, or business men on recreation trips, when the distance to be covered is too long for automobiles to be used advantageously. Many airplanes can be purchased, he said, at prices ranging from $2,500 to $3,500. As they can operate from 15 to 20 miles per gal., the operating expense, including maintenance and repairs, is about $0.03 per mile, or $0.01 per passenger-mile. They operate with a high degree of safety. Although not capable of withstanding so severe a storm as a bombing airplane, they may be compared to a canoe propelled through rapids by an Indian. Weighing only about 800 lb., they can be easily handled on the ground and can carry three persons through the air at the rate of 100 m.p.h. The powerplant of those made by his company consists of a 100-hp. radial air-cooled five-cylinder engine weighing 210 lb. As the cost of an airplane depends mainly on the quantity produced, a very large increase in the production would not be necessary to cause a considerable reduction in the present prices; but a $500 plane will come only when the production is enormous.

Economical Production of All-Metal Airplanes and Seaplanes

By ADOLPH ROHRBACH[1]

AERONAUTIC MEETING PAPER *Illustrated with* DRAWINGS AND PHOTOGRAPHS

ABSTRACT

REDUCTION of cost and of the time required to construct airplanes and seaplanes by applying so-called shipbuilding practice to their fabrication, embodying late types of production methods, are discussed by the author, who says that the company he represents adheres to a number of technical principles to reduce to the minimum the risk of designing and constructing new types. The technical principles refer to general arrangement and to layout, as well as to the detail design of many parts of the planes. They include also very careful and minute preparation for the actual workshop construction by the supplying of perfect workshop-drawings and by proper organization of the technical departments. The paper outlines the technical principles, including reasons for their adoption, and then describes the organization of the work of construction.

Wing-loading and power-loading are discussed, and the statement is made that the company builds monoplanes only. The reasons therefor are explained and the considerations that influence the determination of the right type of flying-boat system are presented. The advantages and disadvantages of side-by-side propellers and of central-tandem propellers are stated and wing construction is described, together with the design of the hull for flying-boats. The differences in airplane compared with seaplane construction are outlined, and details relating to the drawings, jigs and fixtures are included. In regard to the time required for the production of a new type, a flying-boat for which the design was begun about Feb. 15, 1926, was ready to fly in the last days of June, 1926.

IN the present period of airplane development, it seems more important to reduce the time and the cost of production than to concentrate on too much further perfection of constructional details with regard to performance. We possess already sufficient structural and aerodynamical knowledge to be able to construct very efficient airplanes of any size and for every purpose. The methods of stress calculation and performance computation are in most cases more accurate than the basic conditions of such calculations. For instance, we are able to predetermine the factor of safety of a wing within a margin of not more than 5 per cent, although we may be somewhat doubtful whether the factor of safety should be 8 or 10. That is an uncertainty of 25 per cent. A very similar condition exists with regard to the aerodynamic conditions of our performance computation, if based on model tests, and a sufficient amount of full-scale experience is correct within from 3 to 5 per cent; but the actual service-performance of the plane, or that of several planes of the same type, varies much more than 5 per cent owing to weather conditions, to different pilots and the like. Therefore, a much greater amount of work and of capital is needed to improve the performance of a good airplane by 10 per cent than is needed to reduce its cost of production by 10 per cent. So, I think the general use of the airplane, and consequently the chance to sell more of them, will be increased more by a reduction of price of 10 per cent than by an improvement of the same amount. Though this principle could be applied to the airplane industry of any country, it is perhaps most important with us in Germany because we have no military aviation. Therefore, our airplane constructors are forced to sell as many planes as possible in foreign markets. It is difficult for many of these foreign customers to compare the performance of different planes but very easy to compare their prices.

I think that the endeavor to reduce the price does not mean less progress with regard to better quality and performance. On the contrary, if more planes can be sold to the different countries, we get more correct experience out of these planes and can incorporate more improvements in the types that follow. Since, up to now, no market exists for any uniform type of plane, we must be able to construct as many types as possible for every special requirement and to equip them with any desired engine in the shortest possible time. Naturally, the production could be made much cheaper still if a greater number of the same type of plane could be manufactured, as may be possible in the near future, due to the different conditions in your Country. I shall explain in this paper how we, in Germany, endeavor to reduce the time and the cost of building new types.

TECHNICAL PRINCIPLES INVOLVED

The first and perhaps the most important means to achieve the foregoing objective is that we adhere to a number of technical principles to reduce to the minimum the risk of designing and constructing new types. These technical principles refer to the general arrangement and to layout, as well as to the detail design of many parts of the planes. It would be ideal, for example, if the principle which we follow with the construction of wings should prove to be the right one; that is, the type of construction which will dominate as soon as airplanes designed by different constructors are so defined and consequently so similar as automobiles are today. The other principal means to reduce the risk, the construction time and the cost of new types is a very careful and minute preparation for the actual workshop construction by supplying perfect workshop-drawings and by proper organization of the technical departments. I shall describe some of our technical principles first, giving reasons why they were adopted, and then give some idea of the organization of the work. Concerning the latter, it has been developed according to our needs and is constantly being adapted to meet the alterations of our work, so that what I can describe is only a sort of snapshot with a background of principles which are the same in most other companies.

[1] Rohrbach Metall-Flugzeugbau, Berlin, Germany.

TABLE 1—INFLUENCE OF WING-LOADING ON WEIGHT AND PERFORMANCE

	Airplane		
Details	*A*	*B*	*C*
Span, ft.	123	99	87
Wing-Loading, lb. per sq. ft.	9.2	13.5	16.4
Power-Loading, lb. per hp.	21.4	20.4	19.0
Flying-Structure Weight, lb.	7,000	6,050	5,470
Flying Structure, per cent	35.3	32.2	31.1
Powerplant, per cent	18.9	19.8	20.9
Useful Load, lb.	9,100	9,100	8.450
Useful Load, per cent	45.8	48.0	48.0
Maximum Speed, m.p.h.	117	125	132
Landing-Speed, m.p.h.	47	56	62
Service Ceiling, ft.	14,100	9,850	9,200
Range at Full Speed, miles	1,930	2,070	1,990

Wing-Loading and Power-Loading

We endeavor to use the highest possible wing-loading and, correspondingly, to secure the highest power-loading. As is generally known, wing-loading means smaller bending and twisting moments of the wing, a shorter tail and smaller tailplanes and consequently less deadweight, greater strength and endurance of the whole structure and lower building cost. The disadvantages of small wings are reduced ability to climb and higher take-off speed. To a certain extent both these disadvantages can be counterbalanced by a lower power-loading. Another drawback of too high a wing-loading is a relatively bad *L/D* or gross lift-thrust ratio of the whole airplane, because the parasite resistance cannot be reduced below a certain limit and therefore this resistance becomes more and more predominant as the wing area is made smaller. Taking all this into account, it is a matter of compromise to select the best wing-loading for any special purpose. Everyone must make this compromise, but most constructors try to obtain the lowest possible landing-speed so as to have the advantage of greater maximum speed and lessen the influence of gusts. Neither with flying-boats nor with land airplanes have we had any bad experience from their comparatively very high wing-loading.

Table 1 shows, for example, the influence of alterations of the wing-loading in the case of a single-engine night-

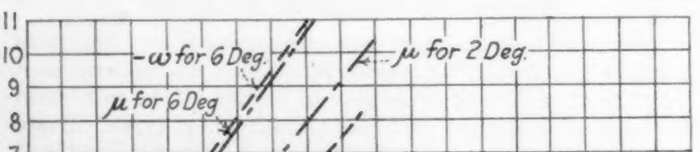


FIG. 1—INFLUENCE OF DIHEDRAL ANGLE ON CONTROLLABILITY
The Curves Show Clearly That the Curve of Controllability Is Increased Greatly If the Wings Are Given a Sufficient Dihedral Angle. They Are for the Dihedral Angles of 2 and 6 Deg. and Give the Time Which Must Elapse After a Rudder Setting of 15 Deg. and a 2-Deg. Aileron-Setting Before a Certain Banking-Angle, a Certain Angular-Velocity and a Certain Deviation from the Original Straight Course Have Been Reached

bombing airplane. In the case of this airplane, high speed and long range were more important than a high ceiling. Even in this case, where climb was not very important, it is useless to increase the wing-loading beyond 14 lb. per sq. ft.

Monoplane or Biplane

We build no more biplanes but build monoplanes only for all purposes. The art of constructing biplanes is relatively much more developed than that for the monoplane. Nevertheless, we have already obtained with the monoplane at least as good and often a much better per-

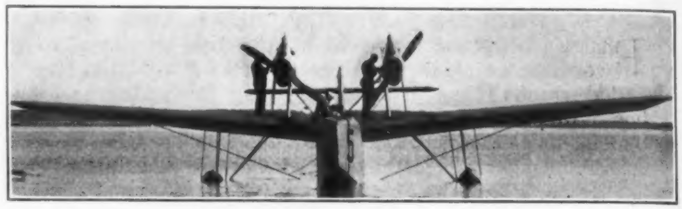


FIG. 2—FLYING-BOAT RO VII, ROBBE
The Bow of the Side Floats Must Be Sharp, As Well as That of the Boat; Otherwise, Too Much Spray Would Be Thrown into the Boat When Taxiing across the Wind

formance than with the biplane. Apart from that the monoplane is more simple, strong and durable and therefore cheaper in every respect. Within a few years, there will be no chance for biplanes in competition with the then so much more refined monoplanes. I have often met people who argued that monoplanes must be less maneuverable than biplanes on account of their greater span. It is true that the moment of inertia of the plane is increased by the greater span, but the monoplane tail is longer, so that the lever arms of all control-surfaces are increased correspondingly. Apart from that, we have

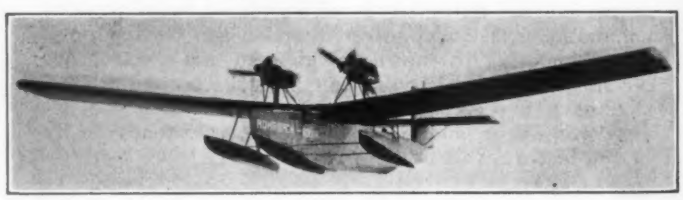


FIG. 3—SIDE-FLOAT ARRANGEMENT
The Construction Illustrated Is That Employed in the Ro IIIa or Rodra Flying-Boat

found that the amount of dihedral angle has much more influence on the lateral controllability than the moment of inertia.

The curves of Fig. 1 show clearly that the curve of controllability is greatly increased if the wings are given a sufficient dihedral angle. The curves presented are for dihedral angles of 2 and 6 deg. and give the time which must elapse after a rudder setting of 15 deg. and a 2-deg. aileron-setting before a certain banking-angle, a certain angular velocity and a certain deviation from the original straight course have been reached. This calculation has been made using the results of numerous model-tests on the theory of small oscillations, according to the theory and under the supervision of Professor Fuchs. The result has been substantiated fully by the flying tests of the flying-boat Ro II, and confirms, for side and lateral controllability, what has long been known for longitudinal controllability. The plane with small longitudinal stability or instability is the most controllable. The high dihedral-angle is the simplest means to give the plane such a very small amount of lateral in-

stability that the highest controllability can be obtained.

In the course of this calculation I have investigated the relative effect of the moment of inertia and of the dihedral angle on side controllability. The result has been that, in the case of the flying-boat Ro II, the effect of 10 per cent increased moment of inertia around the longitudinal axis is counterbalanced by an increase of the dihedral angle of 0.2 deg. Therefore, the influence of the dihedral angle on the flying qualities is much greater than that of the moment of inertia. This perhaps is also an explanation of the experience that planes of the same type sometimes have slightly different flying-qualities, since a slight difference of the dihedral angle may easily escape unobserved, and yet it has a very marked influence on side controllability. These considerations show that there is no special difficulty in obtaining good controllability with a monoplane. The controllability can be really used only if the plane has the capacity of flying

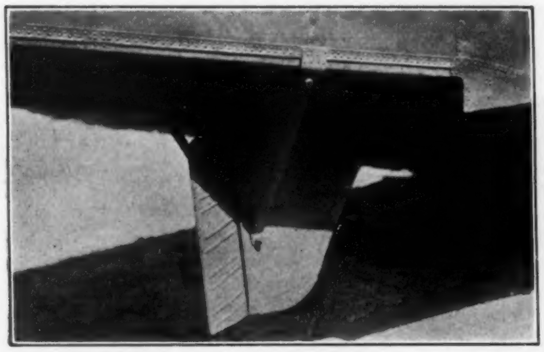


FIG. 4—WATER RUDDER OF FLYING-BOAT RO VII, OR ROBBE
The Desirability of the Water Rudder Is the Same for any Engine Arrangement and Constitutes No Special Disadvantage of the Side-By-Side Propellers

very small circles. The smallest possible circling-radius depends upon the efficiency of the control-surfaces and on the climbing speed of the plane. I repeat that there is no reason why the monoplane should not be better in controllability and maneuverability, as well as in all other performances.

THE RIGHT FLYING-BOAT SYSTEM

Before the construction of our first flying-boat, we made a very extensive investigation regarding the merits of different flotation-systems; that is, of the twin-float arrangement of the flying-boat with small auxiliary-floats underneath the wing-tips of the flying-boat, with sponsons like those of the Dornier Wal and of our system, and with large stabilizing-floats situated relatively near the main hull. We found the last system by far the most promising with regard to floating stability, to weight and to air resistance as well. By making many tank-tests, we succeeded in finding such a shape and position of the floats that they give so much lift during the first half of the take-off period as to permit a considerable reduction of the dimensions of the hull, with a corresponding saving in weight and in air resistance.

The other flotation systems have the following disadvantages: The twin-float system has too much weight and air resistance and insufficient longitudinal stability. The boat with wing-tip floats has no upright floating position at rest and consequently has bad maneuverability at slow speeds in a rough sea. The flying-boat with sponsons has not nearly enough lateral stability because of the excessive extra weight that would be necessary to withstand the water shocks on very large sponsons.

FIG. 5—ENGINE NACELLES
These Are Shown on Flying-Boat Ro III, a Rodra Type. When the Engines Are Located High above the Wing As Shown, They Can Be Placed So Near Together That the Turning Moment of the One-Sided Propeller-Thrust Can Be Counterbalanced Easily Either by a Rudder Setting of About 6 to 12 Deg. or by Setting the Whole Tail-Plane Unit about That Much

The advantages and drawbacks of the different flotation sytems become the more important as the size of the planes is increased. Many pilots expressed serious fears that the stabilizing floats of our plane would break away in a rough sea, but this has not happened yet and not even a strut has been injured. This is due partly to the very high factor of safety of the float struts and partly to the high position of the stabilizing floats in relation to the bottom of the boat, which protects the floats from the heaviest shocks. At high speeds, the boat alone gives all the water lift. It is possible, however, that a side float may be injured or broken off by hitting some floating object or the corner of a landing bridge because of bad maneuvering. In such a case, to protect the plane from capsizing, there are water-tight bulkheads in the side floats and water-tight compartments in the wing-tips. The water-tight bulkheads of the side float proved useful in several cases, when the riveting was not water-eight, or the water drain-plugs had not been closed. But this extra pressure of lateral stability in the wing-tips has never yet been necessary.

FIG. 6—WING OF THE RO II TYPE
Easy Maintenance and Easy Repair of the Structure Are Facilitated by the Detachable Nose and by the End-Boxes Shown

As shown in Fig. 2, the bow of the side floats must be sharp as well as that of the float; otherwise, too much spray would be thrown into the boat when taxiing across the wind. In this connection, Commander H. C. Richardson, Construction Corps, U. S. N., in his paper on The Trend of Flying-Boat Development,[2] mentions in his conclusions:

> For seagoing airplanes, the monoplane probably will dominate, as the wings are less subject to damage in heavy seas in case of a forced landing. The single-float arrangement with a central float probably will dominate. Sponson types are of inferior seaworthiness.

The side-float arrangement is shown in Fig. 3.

Side-by-Side or Tandem Installation?

We prefer the arrangement of two engines side-by-side, on account of the better maneuverability on the water at slow speeds. With the central-tandem-propeller arrangement as used on the Dornier Wal and some other types, rather too many engine-revolutions, with corresponding high taxiing speed, are necessary to obtain some maneuverability by utilizing the slipstream. It is obvious that the better maneuverability provided by the side-by-side propellers is most useful in heavy seas and in narrowly confined waters. The central-tandem-propeller arrangement has the advantage that the maneuverability does not become worse if one engine stops. With the side-by-side propellers, one can taxi a straight course by using rudder and aileron if there is enough wind and if the course is not too much off the wind. Taxiing with one engine and not much wind or on a course more than 60 deg. off the wind becomes possible by using a water rudder, which is shown in Fig. 4. Such a water rudder also would be necessary to taxi more than 60 deg. off a strong wind with central-tandem propellers, so that the desirability of the water rudder is the same for any engine arrangement and constitutes no special disadvantage of the side-by-side propellers.

If one engine fails in the air it is naturally more easy to follow a straight course with the central-tandem-propeller arrangement; but, if the engines are placed high above the wing, as is true of our flying-boats, as shown in Fig. 5, they can be placed so near together that the turning moment of one-sided propeller-thrust can be counterbalanced easily, either by a rudder setting of about 6 to 12 deg., or by setting the whole tail-plane unit about that much. Setting the entire tail-plane unit was used on the greater number of the flying-boats which we have built. It is slightly more complicated to do this, it is true, but it relieves the pilot of the constant pressure on the side control-lever while flying with one engine. This side-control pressure can be counterbalanced by an adjustable spring, for instance, but the one-sided position of the foot control-lever still remains very inconvenient. The other pro and con arguments which are frequently used, such as, for instance, air resistance, weight and propeller efficiency, in regard to the central-tandem or to the side-by-side propellers, balance each other so nearly that neither of the two systems has a distinct advantage over the other. The side-by-side propellers can both be pushers, which gives much protection from damage due to spray in heavy seas.

Wing Construction

Since more than 8 years ago, we have developed the metal-covered wing. The two main problems are (*a*) light weight as a compromise with great strength and endurance and (*b*) easy maintenance and easy repair of the whole structure.

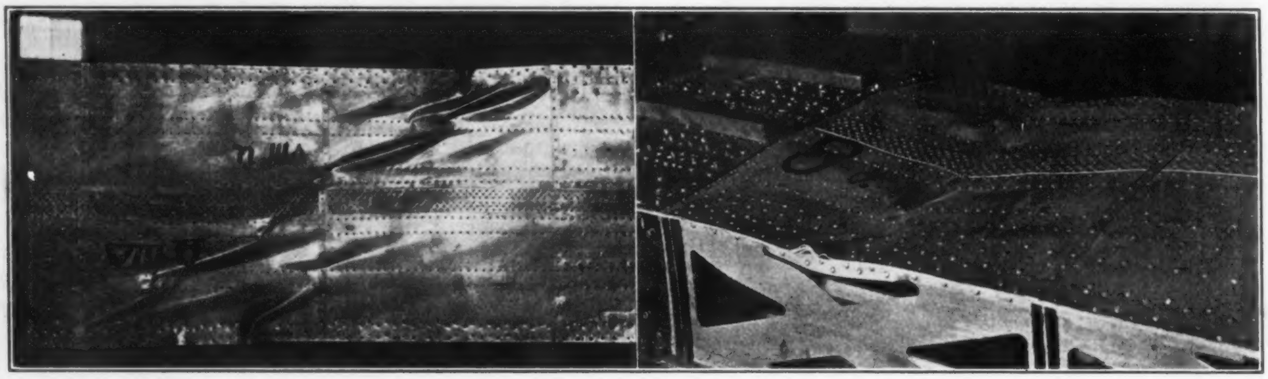


Fig. 7—Box-Girder Tests
The Girder in the Upper View Was Subjected to a Twisting Moment Corresponding to a Very Fast Nose-Drive. That in the Lower View Was Subjected to Bending Stresses Only

The stressed skin made out of flat plates, as used in shipbuilding, is the solution. Task (*b*) is fulfilled by the detachable nose and by end-boxes, as shown in Fig. 6. Though the stressed-skin construction with flat plates is, in principle, most suited to obtain a high and rigid metal-covered wing, a very great amount of detail work was required before this type of wing construction could be mastered. A great number of tests were made since 1919 to determine the best form and arrangement of every detail member of these hollow box-girders.

Fig. 7 pictures a girder test-piece which was broken by bending only and shows another girder destroyed by a twisting moment corresponding to a very fast nose-dive. It was the object of these tests to obtain, as far as possible, uniform strength throughout the girder and to eliminate any unnecessarily heavy part. Through all these tests we have gained so much knowledge that today the strength and the deformation of such girders can be calculated with an accuracy of 5 and of 10 per cent respectively. As we succeeded in getting a lighter and lighter girder of the same strength, naturally the deformation became more and more important. To reduce the torsional wing-deformation during a nose-dive, a wing profile with exceptionally small turning moment of the resultant air-force at zero lift was developed by making aerodynamical calculations and wind-tunnel tests. To reduce the deformation due to bending moments, we

[2] See *Journal of the American Society of Naval Engineers*, May, 1926, p. 16.

Fig. 8—Rupture Test on a Nose-Box
The Strength and the Deformation Can Now Be Calculated With an Accuracy of 5 and of 10 Per Cent Respectively

have adopted a very tapered wing. Fig. 8 pictures a nose-box after testing.

The Boat Hull

The hulls of our first types, Ro II, Ro III and Ro IIIa, have a flat bottom from the first step forward. The bottom from the first to the second step is slightly keeled. The mutual position and shape of the two boat-steps and of the stability floats was developed by making numerous tank-tests. The flat bottom was adopted because it had been the normal German practice with nearly all twin-float seaplanes and with the few flying-boats which we had at the end of the War. From the experience of that time the flat bottom seemed to be the only shape which permitted a short take-off, and the short take-off was very desirable for starts in heavy seas. The first flying boat, Ro II, already had a landing-speed of about 70 m.p.h. Consequently, we designed the bottom to withstand local stresses of about 16 to 20 tons per sq. m.

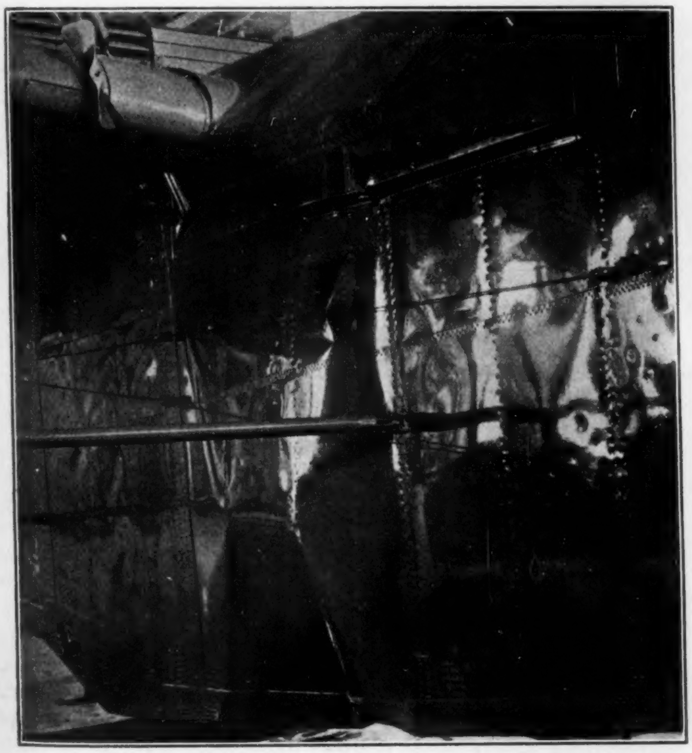


Fig. 9—Flat Bottom of Type Ro III
Two Cases of Damage to the Gliding Bottom Are Evident

(2982 to 3716 lb. per sq. ft.) Trials show that some parts of the bottom are stressed still higher, so that the local strength of some parts had to be increased to 45 and even to 55 tons per sq. m. (8362 to 10,220 lb. per sq. ft.).

Fig. 9 shows a case of severe damage. Type Ro III is our first boat having a keeled bottom. Fig. 10 represents a hull of this type during construction. To find a shape which gives enough lift while gliding on the water with small water-resistance, we had to make numerous tank-tests. The boat seems to have a somewhat longer take-off on calm water and with no wind blowing than does the flat-type Ro III, which took off with two Lorraine Dietrich engines and full contract load in 14 sec. To obtain a short take-off with this type of keeled boat,

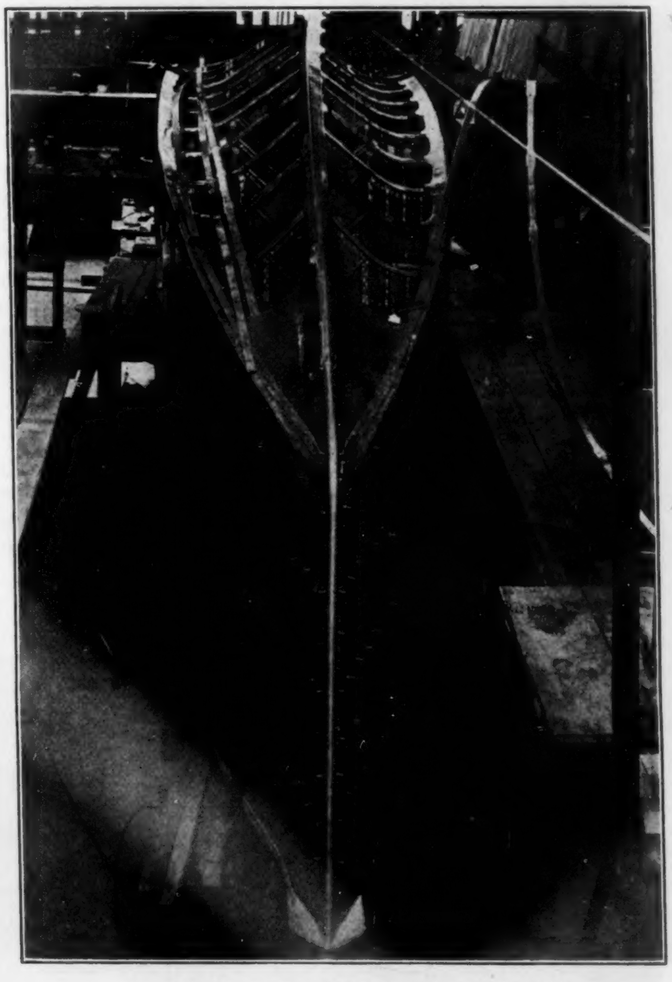


Fig. 10—V-Shaped Bottom of Type Ro VII
Numerous Tank-Tests Were Made To Determine the Shape of the Hull Shown under Construction

the pilot has to know its minimum flying-speed and must draw the machine clear of the water as soon as he knows from the speed indicator that he has reached the minimum flying-speed. The flat-bottom boat Ro III takes-off by itself without the pilot's assistance. In heavy seas, however, the take-off of the keeled-type Ro VII is much smoother and faster. The local stresses on the keeled bottom must be very small, since no bulkhead has ever broken in spite of a very great strength-reduction in the design of these bulkheads with regard to local stresses, as compared with those of the Ro III machine.

Another new and very important feature of the Ro VII type, as compared with the Ro III type, is the very high bow and high deck of the fore part, to obtain greater

FIG. 11—DIFFERENCES IN THE FORE PART OF HULLS
The Ro III Flying-Boat Is Shown at the Right and Two Ro VII Flying-Boats Are Anchored at the Left. The Differences in Hull Shape Are Evident

protection from waves that splash on the deck. Fig. 11 shows our Ro III flying-boat at the right and two Ro VII boats at the left. The difference of the fore part of the hull is visible.

Fig. 12 represents sketches of stiffening bulkheads of the rear part of the Ro II, in the upper view, and of the Ro III in the lower view. The main difference of these bulkheads is that the channel profiles of the Ro II type are connected by two butt-straps on both sides of the profiles and in the Ro III type by one butt-strap between two smaller channel profiles. The later method is much cheaper to build because of the greater accessibility of the rivets.

I need scarcely mention that all boats have water-tight bulkheads arranged so that any two compartments may become flooded without risk of sinking or capsizing. The bulkheads are provided with water-tight doors, equipped with a central quick-closing device which can be operated from either side of the door.

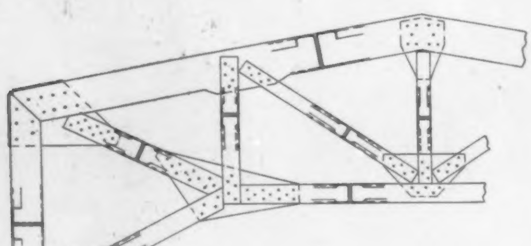


FIG. 12—STIFFENING BULKHEADS
That for the Rear Part of the Ro II Is Shown in the Upper View and for the Ro III, the Type Shown in the Lower View Is Used

PREVENTION OF CORROSION

One of the most important problems is that of corrosion. We had great trouble in this respect but, with all the precautions which we use now, this difficulty seems to be overcome definitely. All profiles and connecting points must be accessible from both sides, so that they can be inspected and repainted easily. All plates, butt-straps and profiles are painted before being riveted together. The rivets are of the same alloy as the plates and profiles and are annealed at 510 deg. cent. (950 deg. fahr.) Rivets that are annealed at a low temperature become corroded much more easily. A kind of black varnish has proved to be the best protective against corrosion. Great care is necessary when other metals are connected with duralumin. Bronze and copper are most dangerous because their electrical tension is very high and this destroys duralumin rapidly. The same difficulty exists with many kinds of chromium steels and also with stainless steel, which contains much nickel. We endeavor to use, against duralumin, steel having a not too high electrical tension and we isolate all steel fittings by thick protective painting between the steel and the duralumin.

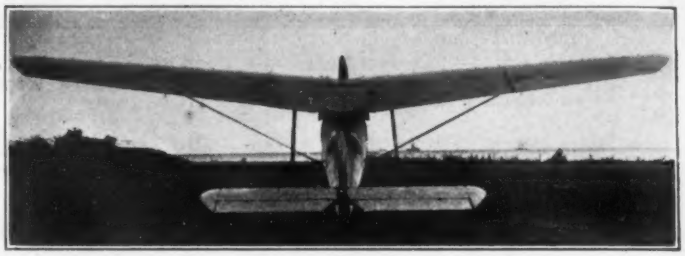


FIG. 13—SINGLE-SEAT COMBAT AIRPLANE ROFIX
The Flying Structure of This Plane Is of the Same Construction, Throughout, As That of the Flying-Boat

AIRPLANES

The first machine of our shipbuilding type of construction was an airplane, namely, the so-called Staaken monoplane having four Maybach engines of 250 hp. each located in the nose of the wing. This machine had to be destroyed in 1920, on account of the Peace Treaty, before much flying experience could be gained. The next airplane was a single-seat combat machine of the Ro IX type having a 460 hp. BMW VI engine, represented in Fig. 13.

The flying structure of this Ro IX plane is of the same construction, throughout, as that of the flying-boat. The landing-gear is of the split-axle type and is sprung by a steel spring which is carried inside to telescope struts. The spring has no initial tension, so the landing is very soft. The wings are braced by cables which have such an initial tension that the bending stresses in the wing girders are reduced to the minimum during flight. The factor of safety is 14.3 for stalling and full load.

Another airplane which is being completed during the writing of this paper is a 10-passenger commercial monoplane with 3 BMW-IV engines of 230 hp. each.

The wing is also braced with the initial tension cables in a manner similar to that for the single-seat plane, and the landing-gear is of the same type of construction. The flying structure is built on the same principles as those used in our other planes. The wing-loading of both these land-airplanes is much less than that of the flying-boats because they must be operated on restricted fields,

whereas flying-boats have unlimited space in which to take-off.

Drawings

To reduce the time and the cost of construction and to use as little material as possible, we endeavor to prepare all drawings, parts-lists and construction orders so that the workman does not need to think about the design, but has only to execute the work as quickly and as well as he can. We do not pay by the hour but by the piece only. The parts-lists are the basis of the orders given to the workshop, and they are arranged so as to comply as completely as possible with the actual construction. For instance, all fittings and bearings of the control-system and of the powerplant installation which are fixed to parts of the wing are mentioned on these wing parts-lists so that they can be fitted during the construction, thus accomplishing great reduction of the assembling time. Every small part and every rivet is specified on some drawing. The drawings are subdivided so that only very few things are represented on any one drawing, to facilitate alterations in the drawings. For the same reason, so far as possible, the dimensions are given on one drawing only. About 1800 drawings and 700 parts-lists are necessary for a flying-boat of the Ro III type.

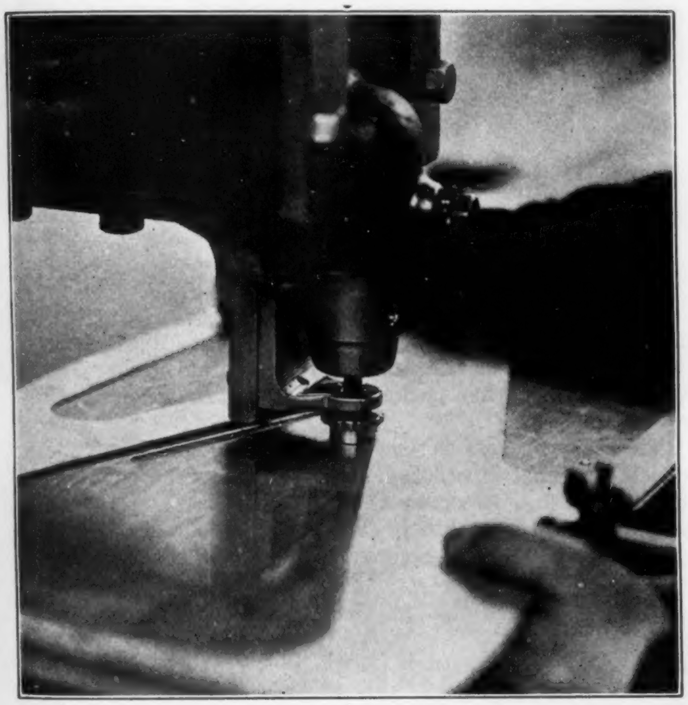


Fig. 14—Cutting the Webs of a Box Girder
The Right Shape of the Cut-Outs Is Obtained by Following a Plywood Jig Which Is Fixed on the Plate

To avoid all unnecessary delay in the execution of workshop drawings, we make a complete initial design before making any workshop drawings. All aerodynamical and strength calculations are made by special departments of the design bureau so that the draftsman has very little to do with such questions, but the initial-design department gives him the thickness of every plate, all profile dimensions, the profile arrangement of every girder, and the number and size of rivets for every important point, as well as the shape of every part which is exposed to the air. This careful preparation minimizes the necessity of alterations and results in such a set of drawings and parts-lists that not only 1 machine but a series of 20 machines could be constructed from them as well. The general style of all drawings is uniform so that less time is lost in the shop before they are understood.

Fig. 15—Jigs Used for Wing-Girder Assembly
The Cut-Out Plates Are Held in the Right Position with the Girder Angle-Profiles by Jigs of the Type Shown

We have made drawings in our office in Berlin for our shops in Berlin and in Copenhagen as well as for our licensees in Japan and in England. The work in Copenhagen, Japan and Glasgow is carried out without any personal supervision from our headquarters office, exclusively on the basis of our drawings and parts-lists. All alterations of the design are specified on the drawings at once. The workshop personnel records all errors and all suggestions for alterations on special forms so that nothing can be forgotten. A complete set of drawings and of parts-lists are kept in the registration office for every individual plane so that, as soon as any certain plane has any interesting experience, we can always check the construction of the respective details.

Jigs

The construction of metal airplanes requires a great number of special tools and jigs. To reduce the number of special tools, we have standardized duralumin profiles, riveting spaces, bolts, and the like. In no other industry is the cheapness of jigs so important as in the aircraft industry, with its frequent alterations of design and its

Fig. 16—Longitudinal-Girder Assembly
The Assembly and Riveting of the Longitudinal Web of a Wing Girder Are Shown. The Transverse Walls Are Made in the Same Way As Are the Longitudinal Girders

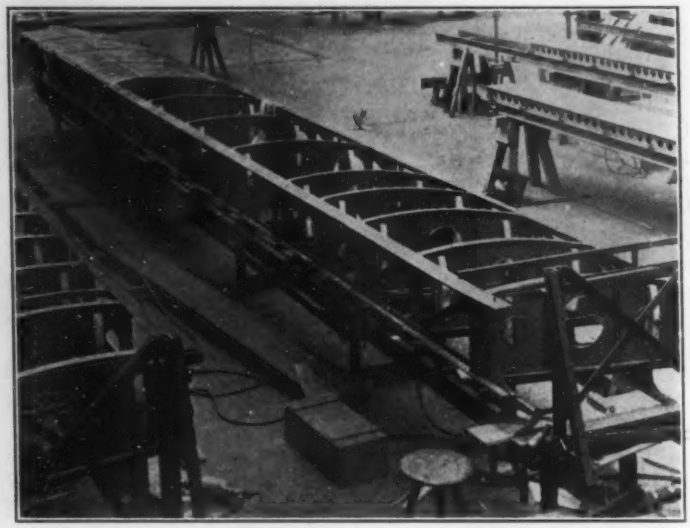


Fig. 17—Wing-Girder Assembly
The Assembling of the Finished Longitudinal Girders with the Transverse Walls Is Done on a Steel Fundament, As Shown

relatively small production. Cheap jigs have the additional advantage that they can be made in a short time, so that the building of a new type of plane is not delayed much by waiting for jigs. It is a special advantage of our shipbuilding type of construction and of our working principle to subdivide the wings into a number of parts so that the jigs are as simple as possible. We have made adjustable jigs for a number of operations, and these can be used for different types of planes.

I have met some designers who argued that, in the case of mass production, a complicated method of construction, requiring expensive tools, may be of advantage over a much simpler type with cheap tools. It is obvious that the cost of jigs becomes less and less important as more planes are produced. This purely economical consideration is true only if the probability of design alterations is very small. At present, the risk exists that a type capable of being produced in great quantities by expensive tools would become obsolete, so that all the capital invested in such tools would be lost. In principle, it is safer and cheaper to use the simplest type of construction and to employ jigs and tools for a few operations only, if a small number of planes is to be built, and to develop additional jigs and eliminate hand-work by special production methods in more and more operations, as the number of planes to be produced in the series becomes greater. The latter method has the advantage that, if the type of airplane is changed, many structural details will remain similar; hence, the mechanical production methods and many of the adjustable jigs can still be employed.

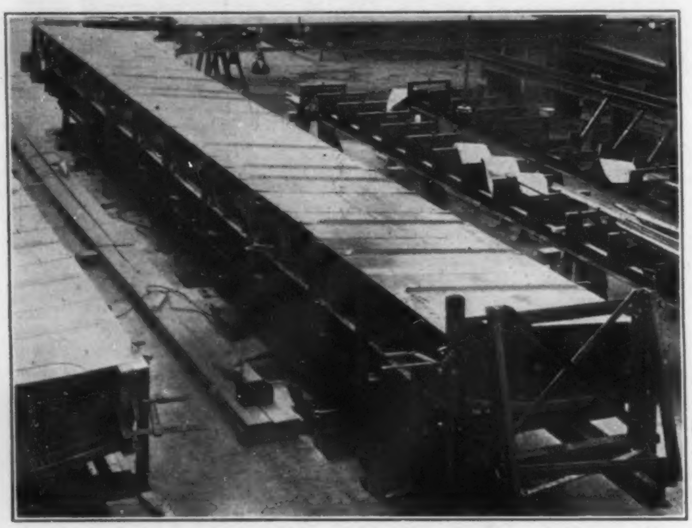


Fig. 18—Installation of the Lower Skin of a Wing
After Installing the Lower Skin, the Girder Is Taken from Its Bed and the Upper Skin Is Riveted On

Wing Construction

The wing consists of the hollow box-girder and nose and trailing end-boxes to complete the wing profile. The webs of the box girder are cut out of plates as shown in Fig. 14. The right shape of the cut-outs is obtained by following a plywood jig which is fixed on the plate. These cut-out plates are then held in the right position with the girder angle-profiles by jigs such as are shown in Fig. 15. On these jigs, the angles are fixed to the girder plates by rivets or screws. The girder is then taken out of the jig and all the riveting is done by riveting-machines, as shown in Fig. 16. The transverse walls are made in the same way as are the longitudinal girders. The assembling of the finished longitudinal girders with the transverse walls is done on a steel fundament, represented in Fig. 17.

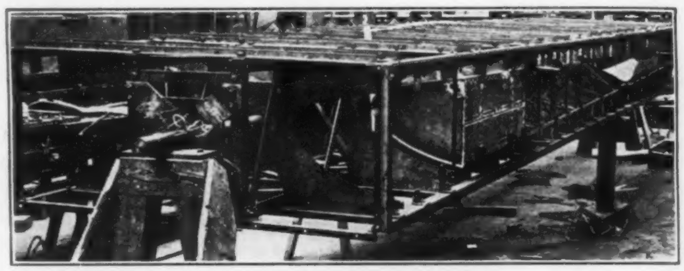


Fig. 19—Jig for a Nose-Box
The Nose End-Boxes Are Assembled Out of Ribs and Skin Plates on a Jig of the Type Shown

The next process is the installation of the lower skin, which was just completed when the view shown in Fig. 18 was taken. After that, the girder is taken from its bed and the upper skin is riveted on. Most of the riveting seams between the separate plates of the upper and lower skin are made by riveting-machines. The assembling rivets which connect the skin with the girders and the transverse walls are riveted with pneumatic hammers. The steel fittings of the joint between the wing and the boat are riveted into the box girder on a special jig, which secures their proper position. All other steel fittings for the attachment of engine or of landing-gear struts, all bearings for the aileron control-rods and all bearings for nose end-boxes are also attached to the box girders, so that it is entirely finished and no more work need be done on this part. The nose end-boxes are assembled out of ribs and skin plates on a jig, as shown in Fig. 19. This jig secures also the right position of the bearings by which they are attached to the box girder later on.

The building of plate-covered control-surfaces formerly required much hand-work and was very expensive. Fig. 20 shows how a simple alteration of the production method eliminated most of this hand-work, the result being that the building-time was reduced by 83 per cent. The left side of the drawing represents the original building-method, where small ribs and the upper and the lower skin were connected by a great number of rivets that, for the most part, were very inaccessible. The

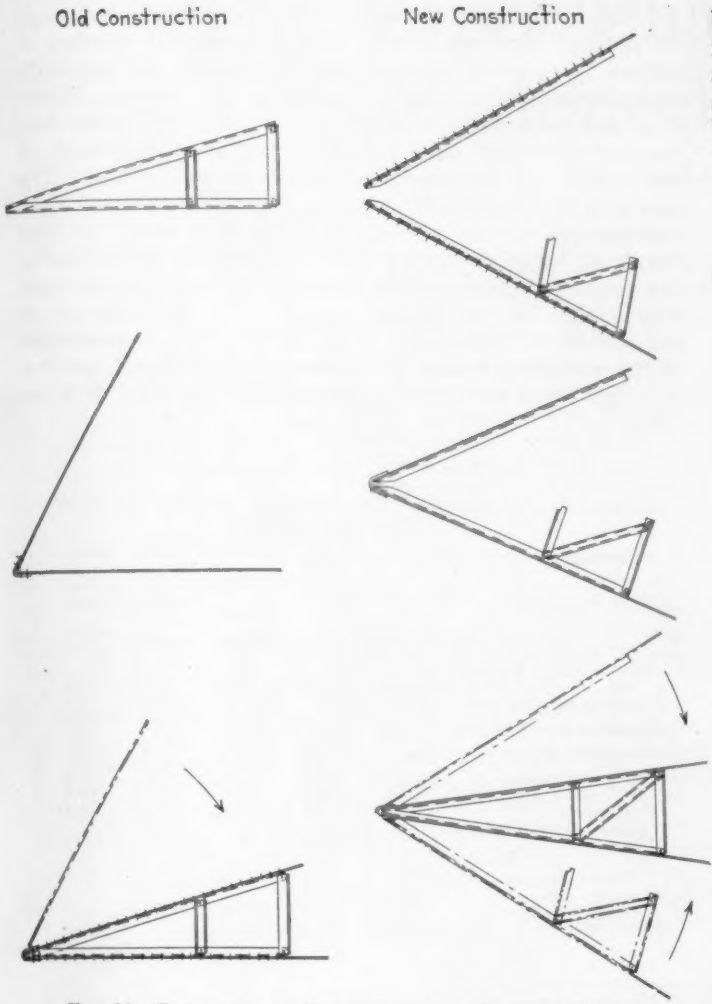


FIG. 20—PRINCIPLES OF CONSTRUCTION OF CONTROL-SURFACE
The Left Side of the Drawing Represents the Original Building-Method in Which Small Ribs and the Upper and the Lower Skin Were Connected by a Great Number of Rivets That for the Most Part Were Very Inaccessible. The Cheap Method Is Indicated by the Drawing on the Right Side

cheap method is indicated by the drawings on the right side. First, the rib girders are riveted to the skin plates on riveting-machines. Then the two skins are connected by the trailing angle, also on riveting-machines. Last, a few rivets are riveted by hand to secure the upper rib-angle to the posts and the diagonals of the rib. Fig. 21 shows the gasoline tanks just before they are enclosed by the end-plates. One of them is placed in front of and the other behind the box girder.

TIME REQUIRED

We have succeeded in reducing the working-time from plane to plane by improving both the design and the production methods. Table 2 shows the working-times for the wing structure of two planes following each other. The working-times of all operations are compared in a similar way.

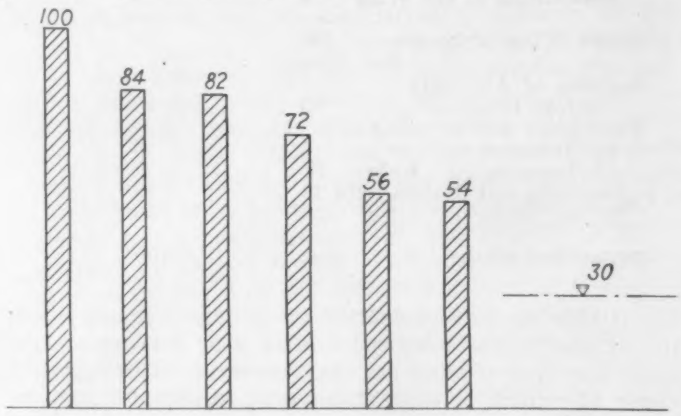


FIG. 22—COMPARISON OF THE BUILDING-TIMES FOR SIX FLYING-BOATS
The Chart Shows How the Working-Time Could Be Reduced if Six Flying-Boats of the Same Type Were To Follow Each Other in Being Constructed

Our subdivided wings, easily accessible for inspection and repair, are at present perhaps 10 or 20 per cent more costly to build than metal-frame wings with canvas covering. Very probably they will be as cheap as metal-frame canvas-covered wings within a short time; but, even if we should not succeed in a further cost-reduction, our type of wing can compete with the metal-frame canvas-wing because the canvas covering must be replaced at least once a year at a cost of about $4 to $5 per sq. m. (10.76 sq. ft.) of surface with us in Germany. This

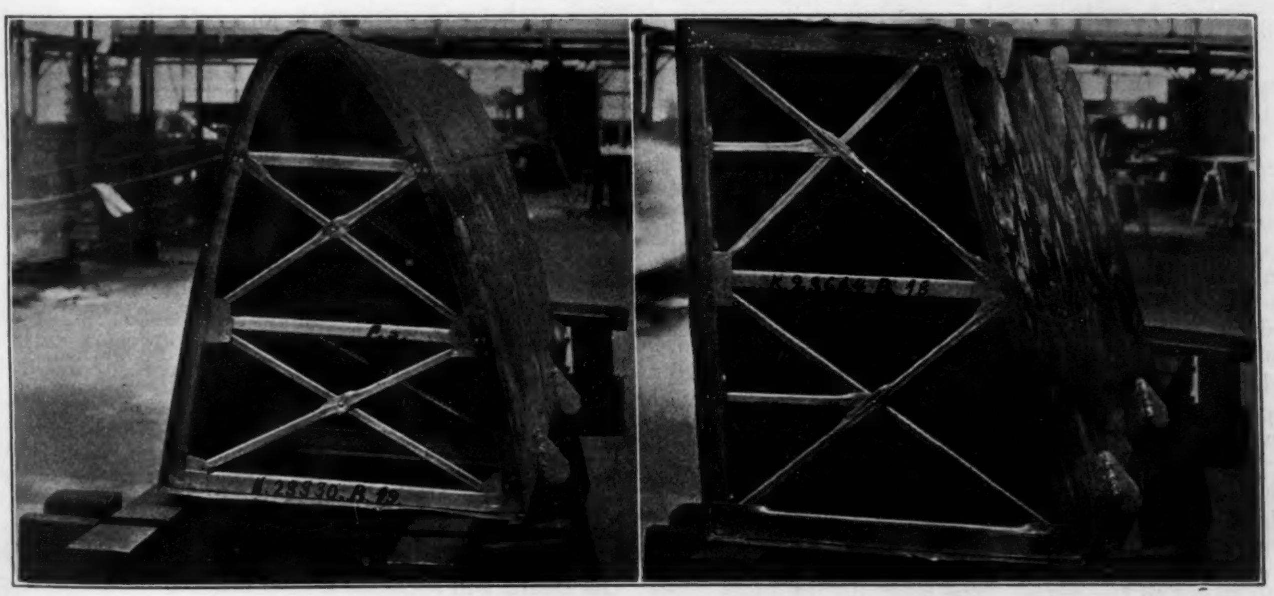


FIG. 21—GASOLINE TANKS
One Is Placed in Front of and the Other behind the Box Girder. They Are Shown Just Before Being Enclosed by the End-Plates

TABLE 2—WORKING-TIMES FOR WING STRUCTURE AND BOX GIRDERS

	Airplane	
Details	A, Per Cent	B, Per Cent
Wing Structure		
Building of		
Box Girder	51	55
Nose Rib-Boxes	10	11
End Rib-Boxes	7	7
Aileron Rib-Boxes	10	9
Aileron	18	6
Wing-Cover	..	8
Assembling of the Wing	4	4
Entire Wing-Structure	100	100
Box Girder		
Building of All Parts except Riveting	30	27
Assembling and Riveting of		
All Transverse Webs	3	5
All Longitudinal Webs	13	8
Top Skin and Bottom Skin	28	30
Entire Girder	26	30
Entire Box-Girder	100	100

means that the replacement of the canvas will cost about 10 per cent of the wing price after 3 or 4 years of service. Our type of wing has the advantage of being much easier to repair. I have endeavored to describe our design and construction principles and to prove that they are developed today so that they can compete very favorably with any other type of construction.

Table 3 gives the main characteristics of several of our later types and shows that very good results are obtained with our type of construction both aerodynamically and structurally, as is indicated by the factors of safety and structural-weight percentages shown in Table 3. Apart from this purely technical viewpoint, I have endeavored to explain that the building cost of all-metal planes can be reduced very materially, even if no great number of planes is ordered, and without impairing the elasticity of a company's procedure regarding the constant alteration and improvement of its types. Fig. 22 shows how the working-time could be reduced if six flying-boats of the same type followed each other in production. The mark on the right side indicates the working-time which corresponds to our present building-methods. Another result of these designing and construction principles is the very short time needed in which to build a new type. The design of our flying-boat Ro VII was started in the middle of February, 1926. The actual construction in the workshop began in the first days of April, and the first two boats were ready to fly in the last days of June, 1926.

TABLE 3—CHARACTERISTICS OF THE LATEST TYPE OF ROHRBACH AIRCRAFT

Designation	Ro IIIa	Ro VII	Ro VIII	Ro IX	Ro V
Type of Craft	Flying-Boat	Flying-Boat	Bombing Land-Airplane	Single-Seat Combat Airplane	Flying-Boat
Type of Engine	Loraine-Dietrich	BMW	BMW	BMW	Rolls-Royce
Size of Each Engine, hp.	450	230	230	460	650
Number of Engines	Two	Two	Three	One	Two
Number of Cylinders	....	Four	Four	Six	Condor III
Maximum Weight, kg.	5,800	3,300	5,700	1,950	9,000
lb.	12,787	7,275	12,566	4,299	19,842
Load, kg. per sq. m.	79.0	82.5	65.0	69.6	95.7
lb. per sq. ft.	16.2	16.9	13.3	14.2	19.6
L/D Ratio	10.7	10.0	12.3	11.0	10.4
Factor of Safety "A Fall"	5.0	4.7	4.6	14.3	5.0
Weight of Flying Structure, per cent	39.7	30.9	30.6	32.3	34.4
Weight of Power-plant, per cent	26.0	29.4	26.6	37.0	22.7
Load, per cent	34.3	39.7	42.8	30.7	42.9

CHRONICLE AND COMMENT

(*Concluded from p. 2*)

Service and Transportation Meeting in Boston in November. Copies of the proposals were sent to the members of the Motorcoach Division and a meeting of the Division was held at New York City on Dec. 20, 1926, for the purpose of formulating recommendations relative to the proposed regulations. The Committee of the Board held a public hearing at Newark, N. J,. on Dec. 21 to receive evidence bearing on the proposals and recommendations as to modifications in them. Representatives of the Division appeared at this hearing. A committee that was organized under the auspices of the National Automobile Chamber of Commerce to develop a code of uniform motorcoach regulations for use as a basis for regulations in all States also submitted recommendations. This Committee has not, however, completed its codification work.

One of the features of the proposed revision of the New Jersey regulations is their separation into two parts, one applying to the "street-car" or "city-type single-deck" motorcoaches and the other to the "de luxe" or "parlor-car" type, not including the "sedan" type. The current regulations that were adopted by the New Jersey Board in December, 1924, have applied to all types of motorcoach operating in the State of New Jersey.

Some of the principal changes proposed are to decrease the minimum inside head-clearance in the parlor-car type from 6 ft. 4 in. to 5 ft. 6 in., omit the guard-rail and glass-partition requirements for the parlor-car type, and eliminate the provision for grab-handles for standees in the parlor-car type. Under "Route Signs," the word "destination" is substituted for "route" for the parlor-car type. In figuring overhang of the body, the total length of the vehicle is to be measured from the front of the radiator to the rear of the body. Some of the most notable changes relate to the height of chassis frame, the change being from 35 to 32 in. for the city type, the specifying of the minimum tread of 72 in. and making the maximum height of the frame for the parlor-car type 30 in. The distance the body may extend beyond the chassis frame has been increased from 10 to 15 in.

H. C. Eddy, chairman of the New Jersey Board's committee for revision of the regulations, stated at the conference in Newark that the comments and recommendations submitted to the committee would be given careful study in preparing the final draft of regulations that will be considered by the New Jersey Board and probably the Board will schedule a final hearing. The Society has been asked to continue its cooperation with the New Jersey Board and its committee in this matter.

The Vapor-Phase Phase of the Antiknock Problem

By W. G. Leamon[1]

Washington Section Paper *Illustrated with* Diagrams and Photographs

ABSTRACT

ALL petroleum-oil cracking operations that have been developed to meet the quantity demand for engine fuel are liquid-phase processes, but the increasing demand for quality is now affording refiners an opportunity to develop distinctive products. In the opinion of the author, cracking processes of the future will be of the vapor-phase type, which gives a product far superior in its antiknock properties.

The composition and distinguishing characteristics of the various families of petroleum oils are explained, including such groups as the paraffins, olefins, naphthenes, and aromatics, the last named including the naphthalenes. The products of liquid-phase cracking are said to contain varying amounts of olefins, naphthenes and aromatics, especially olefins, but all are inferior in antiknock value to California distillate of the same volatility and same distillation range; the product of vapor-phase cracking contains very much higher percentages of these three members, giving it, consequently, a very much higher antiknock value. A particular vapor-phase product, Stellarene, consisting almost entirely of the two series, olefin and naphthene, and produced by the breaking up of large molecules in the vapor phase in the presence of a catalyst, is then described in detail.

THE so-called "cracking" of petroleum oils for purposes of manufacturing automobile fuel is merely a matter of "making little ones out of big ones." The demand of the automobile for increased quantity of fuel has afforded the refiner an opportunity to grow, not only in point of volume of business, but in the equipment for and the means and methods of producing his wares. A modern refinery of today does not even look like a modern plant of 10 years ago. The quality demand of the automobilist now affords the refiner an opportunity to lift his business above the mire of unlimited competition, and protect his major product with the genuine good-will of the buying public for a genuinely good article.

Cracking operations that have become of commercial importance up to the present time have been justified by, and their right to exist has been measured in terms of, the quantity demand for motor fuel. All are liquid-phase processes, that is, oils of large molecule are heated under conditions by which these molecules, supposedly and at least in great part in liquid phase, are shattered or broken up into smaller molecules, the maximum possible percentage of the fragments being made of appropriate size for use as fuel in automobile engines. The conditions are fundamentally only one, namely, the imposition of sufficient pressure on the oils being cracked to raise their boiling-temperature above the temperature of decomposition. I claim some laurels for the petroleum industry for having, by the use of these processes, met the quantity demand for motor fuel in what seems to me to be an economically satisfactory manner, all criticism, just and unjust, to the contrary notwithstanding. I believe, however, and believe it so thoroughly that I have dedicated a considerable proportion of my time to its cause, that the cracking process of the future will be the vapor phase; that is, the demolishing operation will be carried on with the molecules in a vapor rather than in a liquid phase. The reason is that the latest and wisest demand is for quality, commonly thought of as "antiknock" fuels; and the vapor-phase product is far superior in antiknock properties. The demand for such fuels is to my mind a distinct evidence of progress; and I have reason to be an optimist as regards this progress.

Six years ago, when I was first producing in some quantity a vapor-phase cracked fuel, I could scarcely get an audience of one or two, let alone one of this size, to give my theme attention. I was told on every hand that cost per gallon was the only consideration and, at that time, so far as the refiner was concerned, it was. But times have changed.

The Chemistry of Petroleum Oil

In order that I may be sure you have the same point of view as I, and will arrive at the proper conclusions, I ask that you forget your oil chemistry and your knowledge of petroleum refining and take a course with me in both subjects.

Fig. 1, upper left, is a family photograph. The family's surname is Paraffin. The Christian name of each individual in the family appears at the left and the general assemblage of C's, H's and dashes represents carbon atoms, hydrogen atoms and chemical bonds respectively. Each group represents a molecule of oil and differs from the one above it in the addition of one carbon atom and two hydrogen atoms, and in no other way chemically but, proceeding downward, each member has a progressively higher boiling-point and will operate in an automobile engine at a lower compression-pressure without knocking.

This is not the whole range. Pentane with its five carbon atoms is about as small as can be used in engine fuel but they run on down to one; No. 1 is natural gas. This way they go on down to 32, which is the highest. The carbon atoms do not always occur in straight chains. They can be jumbled up a little bit but we will consider this the initial family because it is of the greatest commercial importance at present.

The family shown in Fig. 1, upper right, is known as the Olefin, has similar names for its children, but differs from the Paraffin family in the absence of two carbon atoms. The end atoms of the Paraffin family were hydrogen; in this case, they are carbon. The empirical family formula shows just twice the number of hydrogen atoms as carbon atoms.

In Fig. 1, lower left, we have the Naphthene family, which has the same empirical formula as the Olefin.

[1] A.S.A.E.—Consulting chemist and engineer, Newark, Ohio.

The only chemical difference is that it is built up with a circular structure. So far as antiknock properties are concerned, the Naphthenes and the Olefines for all practical purposes are identical, having the same antiknock values for equivalent-size molecules. They withstand higher compression without detonating than the Paraffins, sufficient in my judgment for automobile practice, but less than the Aromatics, which are shown in Fig. 1, lower right.

THE AROMATICS

The common member of the Aromatics is benzol. The major difference between benzol and its Naphthene cousin of 6 carbon atoms is that only one hydrogen atom appears with each carbon atom instead of two.

The Aromatic family has the highest antiknock value of all the common hydrocarbon-oil families. Toluol is present more or less in the blending of benzol. At the bottom of the figure I have depicted Naphthalene, a solid aromatic of considerable antiknock value when added to gasoline, but its name frequently causes it to be confused with the family name of Naphthene. It is commonly used in moth-balls.

In Fig. 2 are arranged the six-carbon-atom members of all four families in the order of their antiknock value. Paraffins, at the top, predominate in the gasolines that are normally available on the market. In a few crudes, such as some from California and Borneo, a certain amount of naphthenes and aromatics is present, which makes gasoline from those crudes that is higher in antiknock value than ordinary gasoline.

The product of liquid-phase cracking contains varying

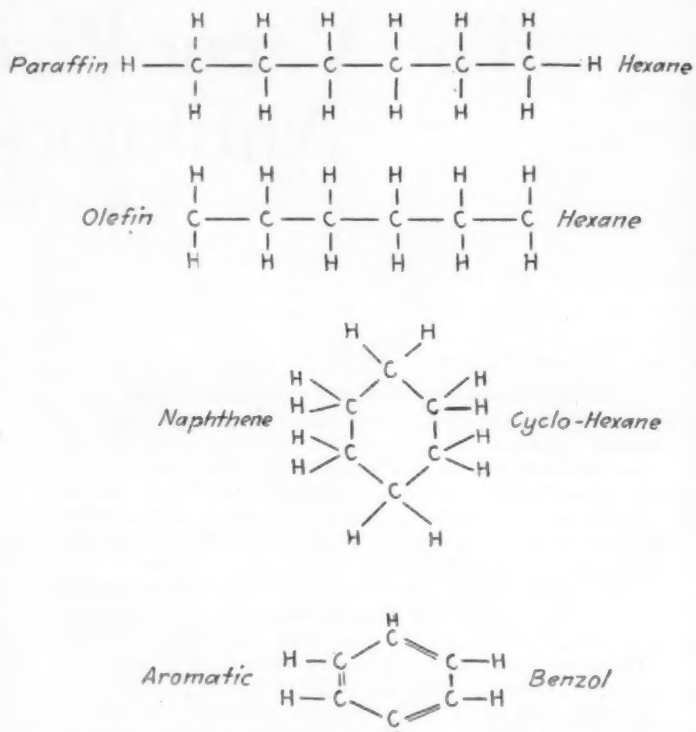


FIG. 2—ARRANGEMENT OF THE SIX-CARBON-ATOM MEMBERS OF ALL FOUR FAMILIES IN THE ORDER OF THEIR ANTIKNOCK VALUE

Paraffins, at the Top, Predominate in the Gasolines That Are Normally Available in the Market. In a Few Crudes, Such as Some from California and Borneo, a Certain Amount of Naphthenes and Aromatics Is Present, Which Makes Those Crudes Higher in Antiknock Value Than Ordinary Gasoline

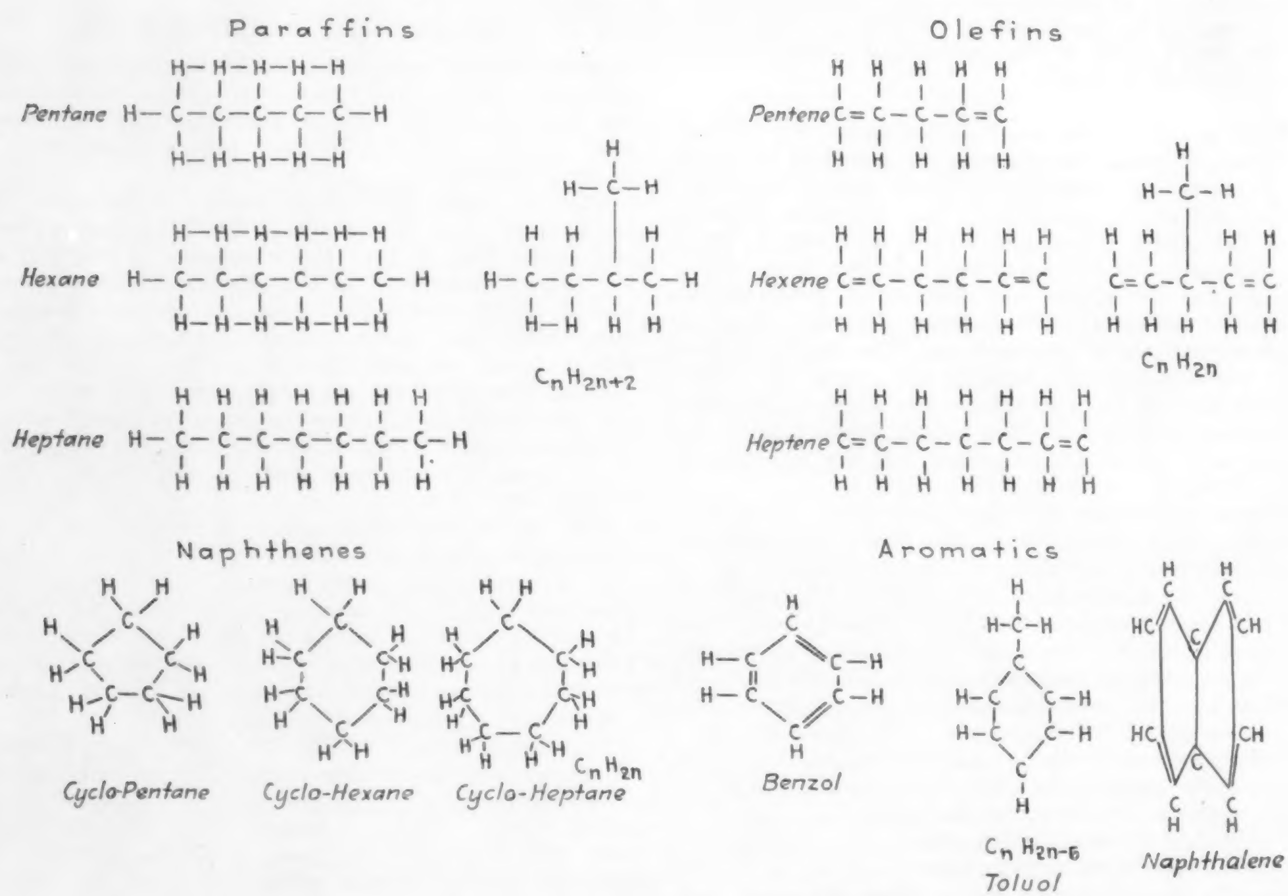


FIG. 1—THE CHEMISTRY OF PETROLEUM OIL

At the Upper Left Is Shown the Composition of a Paraffin Molecule; at the Upper Right, an Olefin; at the Lower Left, a Naphthene; and at the Lower Right, an Aromatic.

amounts of olefins, naphthenes and aromatics, especially olefins, but in no sample of liquid-phase cracked fuels that I have examined have I found the antiknock value to be as high as the usual California distillate of the same volatility and same distillation range.

The product of vapor-phase cracking contains very much higher percentages of these three members, giving it, consequently, a very much higher antiknock value. The vapor-phase product, properly made, is superior in antiknock value to California or Borneo distillates. The vapor-phase product in which I am interested is one in which the breaking up of the large molecules occurs in the vapor phase in the presence of a catalyst. It consists almost entirely of the two series, olefin and naphthene, and bears the name, Stellarene.

Inasmuch as all, or all but a negligible quantity, of the available motor fuels come from crude oil and, in my judgment, will continue to come from crude oil for the next several generations, time will not be badly

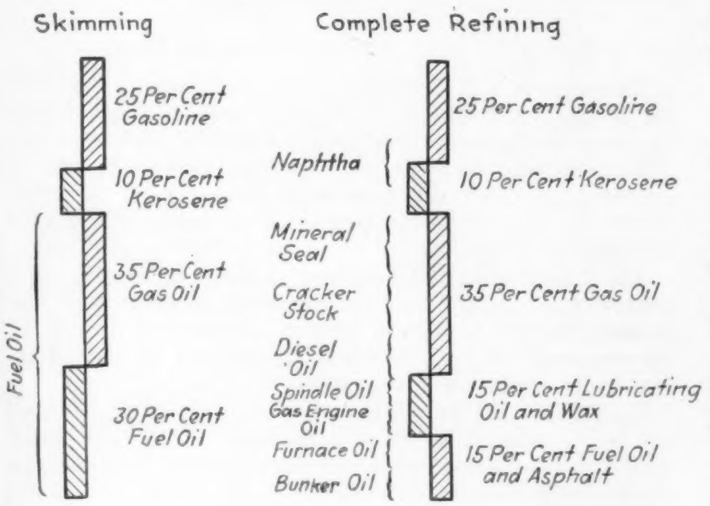


FIG. 3—PRODUCTION OF A GIVEN CRUDE-OIL OF AVERAGE QUALITY BY TWO DIFFERENT REFINING-PLANTS

The Difference in the Boiling-Points of the Different Products Is Taken Advantage Of to Effect Their Separation. In General, the Difference in the Two Operations Is Simply in the Number of Fractions into Which the Original Crude Has Been Cut

spent in looking at the motor-fuel problem from the refiner's position.

The Motor-Fuel Problem from the Refiner's Position

In Fig. 3 is depicted what might be produced from a given crude-oil of average quality by two different refining plants. The difference in the boiling-points of the different products is taken advantage of to effect their separation and, in general, the difference in the two operations is simply in the number of fractions into which the original crude has been cut. The apparatus, the equipment in general, the methods of operation, and the processes employed, however, constitute a very elaborate technique and the difference is considerably more than merely the number of fractions into which it is cut.

The skimming operation can be carried out by a very simple arrangement of apparatus, as is shown in Fig. 4. All the crude is pumped through a heat-exchanger, then through a heater, where its temperature is raised to a point at which all but the heaviest constituents are vaporized. It then passes through an ordinary small tank known as a vaporizer; the heavy oil is dropped and goes into fuel-oil storage; and the vapors leave and go into a fractionating column.

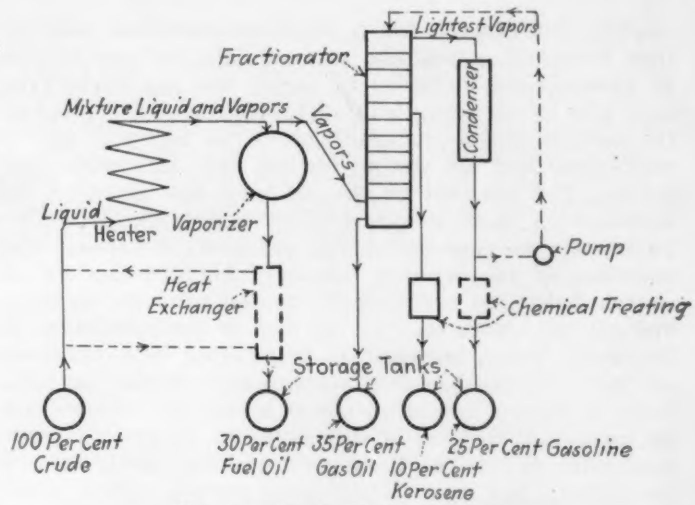


FIG. 4—TYPICAL SKIMMING PLANT

All the Crude Is Pumped through a Heat-Exchanger, Then through a Heater, Where Its Temperature Is Raised to a Point at Which All But the Heaviest Constituents Are Vaporized. It Then Passes through an Ordinary Small Tank Known as the Vaporizer; the Heavy Oil Is Dropped and Goes into Fuel-Oil Storage; and the Vapors Leave and Go into a Fractionating Column

In recent years the petroleum industry has used bubble columns. Nobody can logically explain why it ever used anything else. But the automobile demand for gasoline gave the refiner his chance to grow. The bubble column separates the vapors so that nothing goes over the top except gasoline, and the oils reaching the condenser are gasolines, which may or may not be treated before they go into storage.

At an intermediate point on the column the kerosene may be tapped out and is practically always treated before it goes into storage. At the bottom of the column, every other condensate that occurs is trapped out and goes into what is known as gas oil.

Cleaning Fluid and Burning-Oil

In the early days of the industry, the gasoline market consisted largely of cleaning fluids and the bulk of the production was burned or "accidentally" lost down the river. The real market that supported the industry was kerosene, or burning-oil, for illumination. A limited

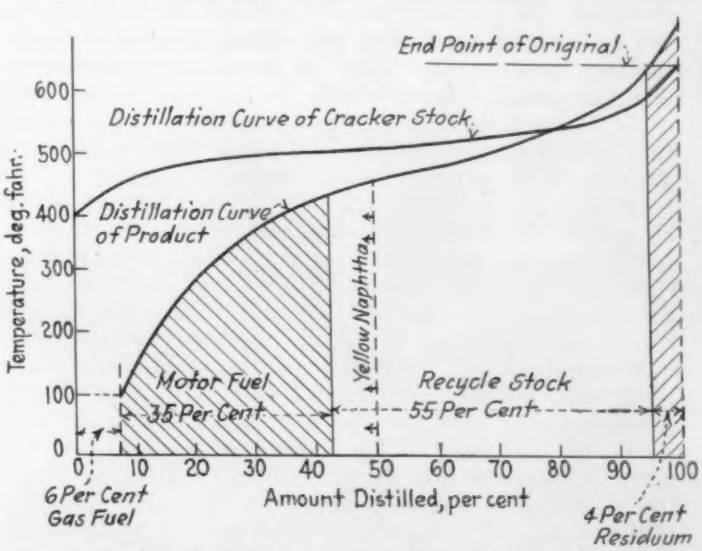


FIG. 5—DISTILLATION CURVE OF AN ORIGINAL CRACKER-STOCK SUBJECTED TO THERMAL DECOMPOSITION

Six Per Cent Is Allowed for the Amount of Gas Fuel That Will Be Produced. This Curve Crosses the Original Curve at about 80 Per Cent of the End-Point of the Original Stock, and Approximately 4 Per Cent of the Product Boils at a Higher Temperature Than the Original. The Motor-Fuel Constituent Is About 35 Per Cent of the Total

market was created for a distillate somewhat heavier than kerosene, which was sprayed into coal-gas retorts or producer-gas furnaces to enrich the gas fuel. This gave rise to the term "gas oil," which is still applied to the portion of the crude oil that is too heavy to use in wick-lamps and not viscous enough for lubricating purposes. The heaviest of the oil was, like gasoline, an abomination to be disposed of as best might be. The automobile is responsible for gasoline's becoming the mainstay of the refining industry, whereas the use of electric lights has curtailed the market for both kerosene and gas oil. More than 80 per cent of the population of the world today, however, is illuminated with kerosene oil, and kerosene has also a substantial market as fuel. It has a further important use as a fuel for tractors and its market, therefore, is still sufficient to sustain, with reasonable profit, the kerosene-producing operations in the refinery, but gas-oil and fuel-oil portions of the crude

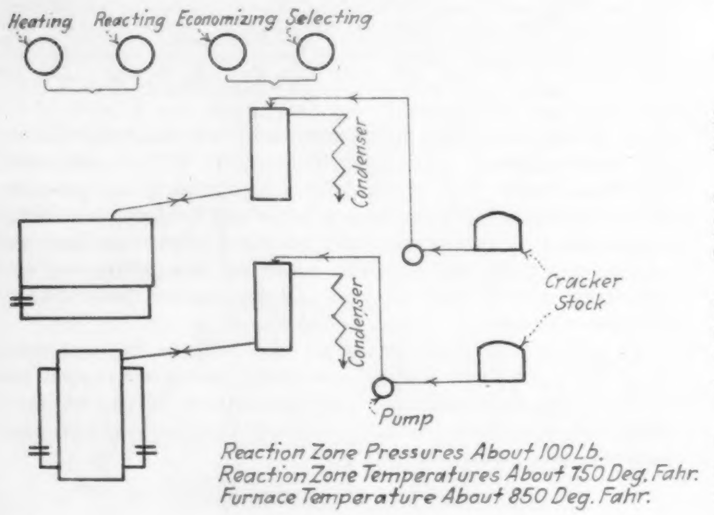


FIG. 6—TYPE A ADAPTATION OF THE HEATING, REACTING, ECONOMIZING AND SELECTING ELEMENTS

The Heating and Reacting Units Are Combined, As Are Also the Economizing and the Selecting Elements. The Cracker Stock Is Fed through the Tower Where It Receives Heat from the Vapors That Have Left the Still, and Flows in This Heated Condition into the Still

are normally handled at a loss. Even lubricants can be handled economically only in large and expensively equipped plants. Economically, therefore, a process that could convert these unprofitable portions of the crude into gasoline would be able to bear a heavy expense if attended with only a moderate degree of success. These portions are, therefore, the raw material about which all present-day cracking processes have been developed.

Fig. 5 shows the distillation curve of a typical cracker stock or oil to be subjected to thermal decomposition, and also the typical distillation-curve of the product that will be obtained after the stock has been thermally decomposed and the various products of decomposition have been brought into equilibrium with each other, allowing 6 per cent for the amount of fixed gas that would be produced. This curve crosses the original curve at approximately 80 per cent, approximately 4 per cent boiling at a higher temperature than the end-point of the original. The motor-fuel constituent is about 35 per cent of the total. It is interesting, if not significant, that the distillation curve of the product is a fair representation of the distillation curve of an average crude-oil as found in nature. This diagram is approximately correct and equally applicable to all processes and all cracker stocks.

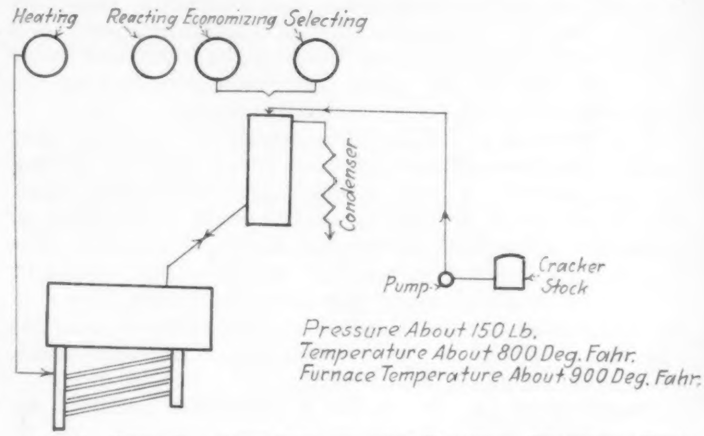


FIG. 7—TYPE B ADAPTATION OF THE HEATING, REACTING, ECONOMIZING AND SELECTING ELEMENTS

The Heating and the Reacting Elements Have Been Separated and Subjected to More Direct Control. The Economizing and the Selecting Systems Are Still Together

ELEMENTS OF A PROPER CRACKING SYSTEM

The elements of a proper cracking system, a system for subjecting the oil to thermal decomposition, are four in number: heating, reacting, economizing, and selecting.

The heating element is very important, because it operates at about the maximum temperature at which petroleum oil can be handled, that is, from 700 to 1100 deg. fahr.

The function of the reacting element is to provide a proper time factor at the cracking temperature employed; and, to a certain extent, within narrow limits, the temperature and the time factors can be varied inversely to achieve a given result.

Because of the very high temperatures employed and the large amount of heat contained in the oil, which should be recovered and not lost in the process, the economizing element is vital.

Inasmuch as approximately only one-third of the cracker stock can be converted into gasoline-size molecules in the reaction zone at one time, and in practice most processes give a much lower yield, it is necessary

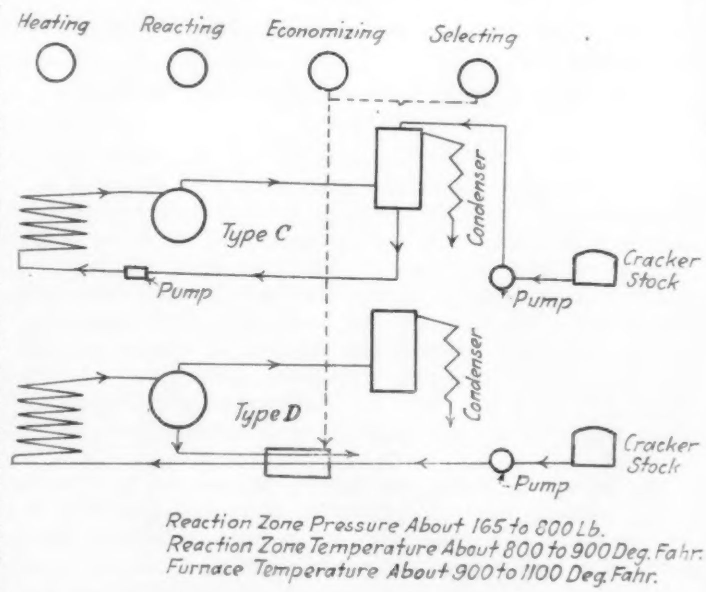


FIG. 8—TYPE D ADAPTATION OF THE HEATING, REACTING, ECONOMIZING AND SELECTING ELEMENTS

The Economizing and the Selecting Elements Are Separated and Heat Is Transferred from the Residual Oil by a Heat Exchange between the Cracker Stock Entering the System and the Residual Oils Coming Out. Various Combinations of All the Types Shown in Figs. 6, 7 and 8 Are Being Used

that a selection be exercised on the outgoing product, so that most of the unconverted molecules can be returned directly or indirectly to the reaction zone.

In Fig. 6, the original design, we have what I have designated as a Type A adaptation of these four units. The heating and reacting units are shown combined, as are also the economizing and the selecting elements. The cracker stock is fed through the tower where it receives heat from the vapors that have left the still, and flows in this heated condition into the still.

In Type B, Fig. 7, which came very soon, the heating and the reacting elements have been separated and subjected to more direct control. It is built very much after the pattern of a water-tube boiler, but the economizing and the selecting systems are still together.

In Type D, Fig. 8, the economizing and the selecting-units are separated and heat is transferred from the residual oil by a heat exchange between the cracker stock entering the system and the residual oils coming out.

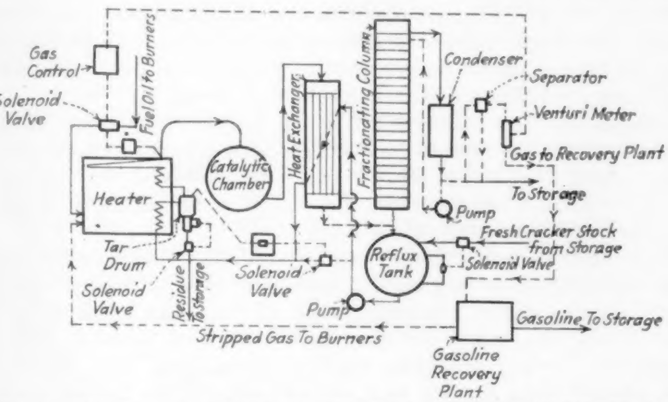


FIG. 9—METHOD OF ASSEMBLING THE FOUR CRACKING ELEMENTS IN THE STELLARENE UNIT

Each Element Is Entirely Separate and Distinct and Is Under Separate Control. The Selecting Is Done by a Bubble Column Similar to That Used in the Alcohol Industry. In the Reacting Element Is Housed the Catalyst That Controls the Composition of the Product and Reduces the Time of the Reaction

Various combinations of the elements of all these types are being used.

Assembling Four Cracking Elements in Stellarene Unit

Fig. 9 shows diagrammatically how the four cracking elements are assembled in the Stellarene unit. Each element is entirely separate and distinct and is under separate control. The selecting is done by a bubble column similar to that used in the alcohol industry. The heat-exchangers are of a vapor-to-liquid type, the design of which was developed for this purpose. The fractions that are condensed in the fractionating column, as well as all the intermediate condensed oils, are returned directly to the heater, together with a supply of fresh cracker stock. The heating element is the well-known Foster oil-heater, with extended surface-rings over seamless-steel tubing. We unequivocally claim the best reacting-element for the reason that in it is housed the catalyst that controls the composition of the product and reduces the time of the reaction.

Fig. 10 shows a more or less complete layout of a Stellarene unit. All oil returned by the selector, that is, the fractionating column, for reprocessing, as well as all heavy oils intermediately condensed in the heat-exchangers, together with the fresh cracker stock, are mingled in the so-called reflux-tank. The pump takes the oil from this tank and forces it through the economizing units, that is, the heat-exchangers, to the lower heating-coils of the heating element. At a predetermined temperature, the mixture of oil and vapors is passed out of the heater into the so-called tar-drum. Accurate control of this temperature is effected by means of a pyrometric controller connected to a solenoid valve on a by-pass around the heat-exchangers. In the tar-drum, the tars and extremely heavy oils that could not be converted into vapors are dropped out and allowed to escape through a solenoid valve on the escape line, which is operated by a constant-level float-regulator. The vapors pass back into the heating element where they are raised to as high a temperature as is possible without carbonizing the tubes of the heater. When a uniform cracker-stock is used, this temperature can be determined experimentally and maintained by a constant-temperature pyrometric control connected with a solenoid valve on the fuel line to the furnace of the heating element. When the characteristics of the cracker stock vary, a different system is used. The vapors pass from the heater through the reacting zone containing the catalyst, thence to the heat-exchangers and the fractionating column, where all but the desired converted product and the fixed gases produced are condensed and trapped back into the starting-tank.

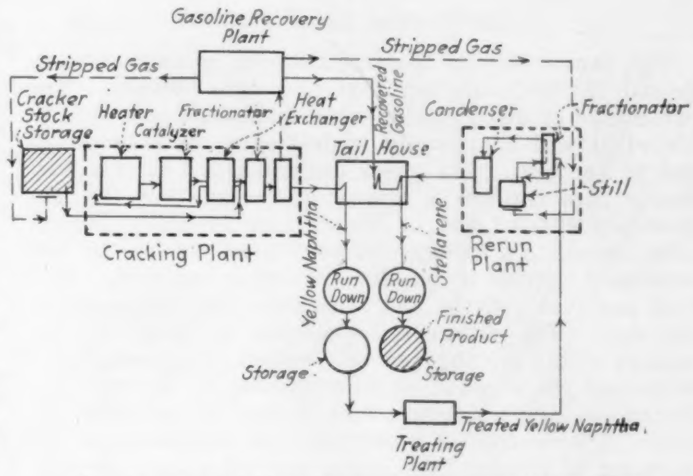


FIG. 10—DIAGRAMMATIC SKETCH OF A STELLARENE UNIT

By Maintaining a Constant Flow of Gas, the Cracking Reaction Is Carried On at a Constant Rate Regardless of the Character of the Cracking Stock, and the Human Element in Determining the Correct Heater-Temperature and Holding to It Has Been Eliminated

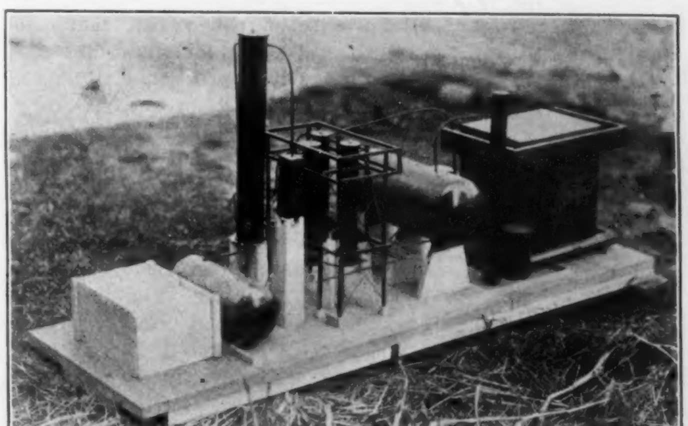


FIG. 11—VIEW OF MODEL STELLARENE UNIT

This Shows the Geometrical Arrangement of the Various Units, Namely, the Heating Element, the Reaction Unit, the Three Heat-Exchangers Constituting the Economizing Element, the Bubble Column or Selecting Unit, the Condenser, and the Tank from Which the Feed Is Obtained To Keep the System Supplied with Oil

Condensing the Vapors

The vapors leaving the fractionating column are condensed in the condenser and flow into storage. The percentage of motor fuel in this condensate can be controlled at will but usually is desired to be between 70 and 90 per cent. The gases not condensed in this condenser pass through a separator, to take out the mechanically carried mist of liquid, then through a venturi tube, thence to a tail-gas recovery-plant, where the last remaining portion of gasoline vapors is removed. This fixed gas then returns and constitutes the fuel used by the unit. The differential pressure produced by the passage of the gas through the venturi tube is employed to control the amount of fuel passing to the furnace. By maintaining a constant flow of gas, the cracking reaction is carried on at a constant rate regardless of the character of the cracking stock, and the human element in determining the correct heater-temperature and holding to it has been eliminated.

Fig. 11 is a photograph of a model Stellarene unit, showing the geometrical relation of the various units, namely, the heating element, the reaction unit, the three heat-exchangers constituting the economizing element, the bubble column or selecting unit, the condenser, and the tank from which the feed is obtained to keep the system supplied with oil. The condensates, from the bottom of the tower and from the bottom of the heat-exchangers flow into this tank and, through the agency of a level regulator, fresh cracker stock is admitted of sufficient amount to keep the level in the tank constant. In the little house at the end of the unit opposite the heater are the pumps and instruments used in controlling the operation.

Method of Operation of a Complete Plant

In a complete cracking-plant, we can follow the course of the oil from the cracker-stock tank through the heating, reacting, economizing and selecting elements, thence to the condenser and, through the glass-enclosed look-box for inspection, to the intermediate storage-tank. The product at this point is a crude cracked product called, in the Stellarene process, "yellow naphtha." Other processes christen their analogous products "synthetic crude," "pressure distillate" and numerous other names. The yellow naphtha is taken from storage, given a treatment and rerun in an equipment somewhat similar in principle to the skimming plant. The motor fuel passing from the fractionating column of the rerun plant is condensed and flows through another glass-enclosed look-box in the tail-house, and thence to the finished storage. The gas leaving the Stellarene unit is taken to a recovery plant, where the last bit of motor fuel is extracted and sent back through the same or an equivalent look-box for inspection, and thence to either the intermediate or the finished-product storage. The stripped gas from the recovery plant is used as fuel. We shall now proceed to acquaint ourselves with this finished product.

It has become an established practice to define the antiknock value of a fuel in terms of the percentage of benzol necessary to be added to a standard gasoline in order to make the blend produce as nearly as possible the same amount of knock in an engine. In other words, the fuel is run under conditions that are observed, and the engine is brought to the point at which it knocks. Then a standard paraffin gasoline is blended with the benzol in varying amounts until there is sufficient benzol in the blend so that it will operate the same, as nearly as can be observed, as the unknown fuel. Obviously, the composition of standard gasoline and the purity of the benzol are controlling factors and account for some of the discrepancies often observed in the work of different laboratories. It has been found that the performance of a fuel in an engine running under conditions of established detonation, that is, continuous knock, is different from its performance when the engine is running out of detonation, that is, without knocking, and that therefore a fuel may, and in fact does have in some cases, two benzol-equivalent values.

Stellarene

Stellarene is such a fuel. It is to be noted that increasing amounts of benzol in the reference blend have an increasing antiknock effect. This is the reverse of the usual effects of the so-called "dopes."

The first 5 per cent of benzol has virtually no antiknock value; any gasoline, therefore, can be said to be almost equivalent to a blend of 5-per cent benzol with itself. A 10-per cent blend is not much better. In Fig. 12, the shaded areas indicate the relative effectiveness of progressive increases in the percentage of benzol in 10-per cent steps. The first step is small, the second is double the first, the third is 50 per cent greater than the second, the fourth 25 per cent greater than the third. In other words, a 40-per cent benzol-blend is 2.8 lb. better than a 30-per cent blend, but a 10-per cent blend is only 0.7 lb. better than none at all.

Stellarene has usually been found to be equivalent to a blend of paraffin-gasoline with from 30 to 40 per cent of benzol. Some determinations of the benzol equivalent of Stellarene, when running at the highest possible compression before inducing regular detonation, have shown a higher than 40-per cent equivalent. This is the condition that will ordinarily be found in automobile practice. Other determinations have been made which show that the benzol equivalent, under conditions in which regular detonation exists in the engine, is less than 30 per cent. Generally speaking, and I believe for all practical purposes so far as automobile use is concerned, Stellarene may be safely considered as equivalent to an ordinary gasoline-benzol blend containing 35 per cent of benzol. The best gage of the value of a cracking process as an antiknock motor-fuel producer is probably the determination of the benzol equivalent of its product, using as standard gasoline the natural gasoline obtained from the same crude oil from which the cracker stock was made.

Tests for Antiknock Value

In Fig. 13 are plotted the amounts of gasoline produced in a Midgley bouncing-pin test-apparatus by various fuels. Without going into detail, it is sufficient to note that the antiknock value is indicated by the displacement of the curves to the right. The first curve is for kerosene. The second is for United States motor gasoline made from an Appalachian field crude and having an end-point of 437. No. 3 and No. 4 are Motor Stellarene made from the same crude, the latter having a 390 end-point. The difference between curves No. 3 and No. 4 shows the effect of volatility on the antiknock value. The drop from the 437 to the 390 end-point had many times the effect of the addition of the last 10 per cent of benzol in a 40-per cent blend. The relation of curve No. 3 to the curves No. 2A and No. 2B defines the antiknock value of this particular sample of Stellarene rather definitely in terms of these two particular benzol-blends, 2A being a 30-per cent benzol and 2B a 40-per cent benzol-blend with curve No. 2 gasoline.

A great deal has been said of late, and a great many prophecies have been uttered, concerning the value of antiknock fuels so that, logically, I feel compelled to express my conception of the reason for and the future of antiknock fuels, and I shall do so at the risk of being charged with heresy. Even the first page of the Babson Report of March 30, 1926, is devoted to The New Motor and the New Motor Fuel, mentioning first the new engine. But I am not expecting to see the oft-referred-to "new" engine, of greatly increased economy and efficiency, due to greatly increased compression-ratio. I believe the present automobile engine is already over-compressed to the point at which the present proposed fuels can only meet present requirements. The average automobile of today, and I have seen many tested, will show a 20-per cent increase in mileage when supplied with a fuel of adequate antiknock properties. This is sufficient economically and mathematically to justify a premium of 20 per cent, which my experience tells me to be just about the psychological limit. But that is not the reason for the popularity of antiknock fuels. Experience again has convinced me that people pay premiums for automobile fuel because of the greater satisfaction they get from driving an engine that, to use the customary explanations given by laymen, "runs more smoothly," "does not knock," "runs more quietly," "accelerates more quickly." "Christian Scientific?" Perhaps; but they are factors and the controlling factors in selling a premium motor-fuel. I would rather sell, because it is easier to sell, 1 oz. of satisfaction than any pound of efficiency or economy that may be devised for the good of the American Public.

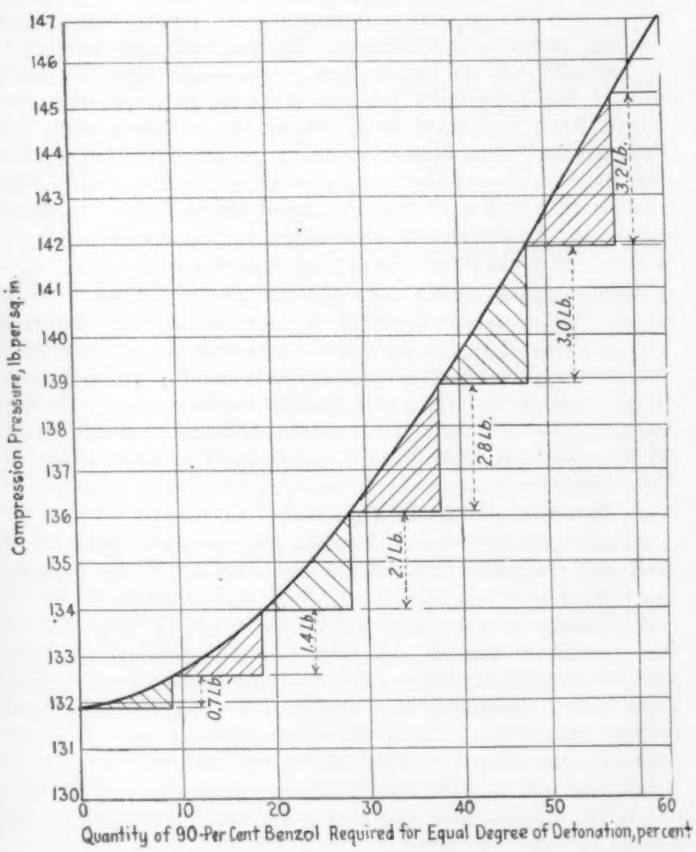


FIG. 12—RELATIVE EFFECTIVENESS OF PROGRESSIVE INCREASES IN THE PERCENTAGE OF BENZOL IN 10-PER CENT STEPS

These Are Indicated by the Shaded Areas. The First Step Is Small, the Second Double the First, the Third 50-Per Cent Greater Than the Second, the Fourth 25-Per Cent Greater Than the Third. In Other Words, a 40-Per Cent Benzol Blend Is 2.8 Lb. Better Than a 30-Per Cent Blend, But a 10-Per Cent Blend Is Only 0.7 Lb. Better Than None at All

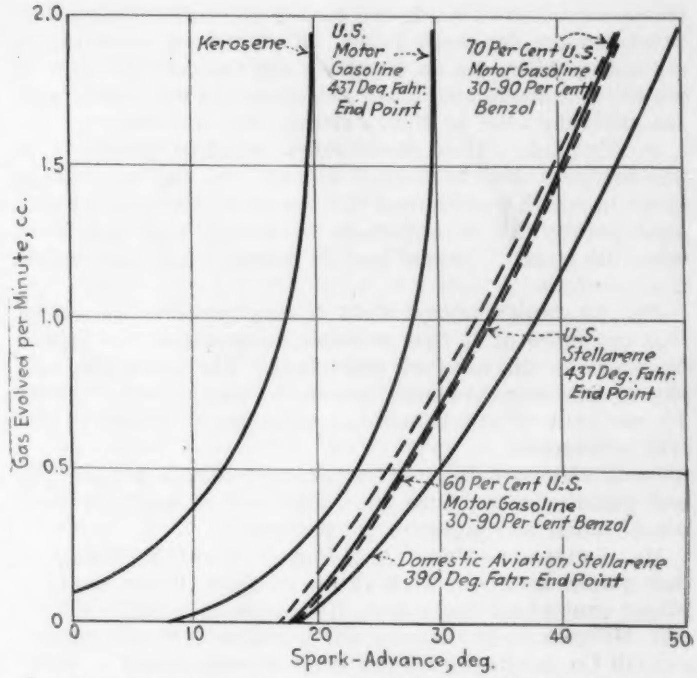


FIG. 13—AMOUNTS OF GASOLINE PRODUCED IN A MIDGLEY BOUNCING-PIN TEST-APPARATUS BY VARIOUS FUELS

The Antiknock Value Is Indicated by the Displacement of the Curves to the Right. The First Curve Is for Kerosene; the Second Is for United States Motor Gasoline Made from an Appalachian Field Crude and Having an End-Point of 437; the Third and the Fourth Are for Motor Stellarene Made from the Same Crude, the Latter Having an End-Point of 390. The Drop from the 437 to the 390 End-Point Had Many Times the Effect of the Addition of the Last 10 Per Cent of Benzol in a 40-Per Cent Blend. The Relation of Curve No. 3 to the Dotted Curves 2a and 2b Defines the Antiknock Value of a Particular Sample of Stellarene in Terms of Two Benzol Blends of United States Motor Gasoline

THE DISCUSSION

CHAIRMAN P. B. LUM[2]:—Did you say that it would not be economical to use shale for some time to come as a source of petroleum oil?

W. G. LEAMON:—I think it will be a great many generations before we shall use shale oil extensively. Shale probably is the logical next source of supply of petroleum oil but, in my judgment, its use is far off.

A MEMBER:—Do you mean that the quantity of oils that will be produced will continue large and that by cracking them you will get a greater quantity for consumption?

MR. LEAMON:—Both. The only reason for needing more crude oil in the future will be to make more motor fuel. The statement was recently made to me that the controlling factor in the petroleum industry would be lubricants. In a completely refined sample of crude oil, lubricants on the average will represent less than 15 per cent. There is no market today for the lubricants that are produced, and lubricants can be produced properly only in very large quantities, in very expensively equipped plants. If there is 5 per cent of lubricants on the average in crude oil, I presume there is 5 per cent of that quantity being made. The rest is going into cracker stock or fuel oil.

The controlling factor is motor fuel. The important increased demand is not for lubricants but for motor fuel, and the amount required will therefore increase the amount of crude necessary, but the amount of crude produced is capable of being enormously increased at the present time.

We are a long way from 1,000,000,000 bbl. a year, and

[2] M.S.A.E.—Manager, Autocar Sales & Service Co., City of Washington.

the present known fields will supply not only 1,000,000,000 bbl. but more for many years. So, I am not expecting to see automobiles run on anything but the fuel that can be made from crude oil. I am not expecting my children to see anything else, or theirs either, for that matter.

A MEMBER:—How does ethyl gasoline compare in percentage with the benzol blend? In the percentage chart in which you showed the average ethyl blend being used before the manufacture of benzol was begun, to what per cent of benzol and of straight gasoline would that compare?

MR. LEAMON:—Ethyl fluid is analogous to benzol in that the more of it that is added to gasoline, the higher compression the mixture will stand. The quantities employed, however, are very minute in comparison. About 0.1 per cent of ethyl fluid is equivalent to 30 or 40 per cent of benzol.

A. W. HERRINGTON[3]:—The amount of ethyl fluid put into gasoline to produce the equivalent of a 35-per cent benzol blend is roughly 5 cc. per gal.

MR. LEAMON:—Yes. A gallon is about 3800 cc., so that would be about 0.13 rather than 0.10 per cent; a 35-per cent blend equivalent, I suppose, is about right.

A MEMBER:—Is the new SO gasoline that the Standard Oil Co. is placing on the market ethyl fluid?

MR. LEAMON:—No. The so-called Standard Antiknock, colored red, which is being distributed in this part of the Country now, contains no ethyl fluid, or did not contain it when first brought out.

A MEMBER:—The name has been changed to E-s-s-o instead of Antiknock.

CHAIRMAN LUM:—Mr. Leamon, did you say that fuel oil, which is 30 or 35 per cent of the total residue, is sold at a loss?

MR. LEAMON:—Yes.

CHAIRMAN LUM:—Is that what you draw upon in this cracking process?

MR. LEAMON:—In all cracking processes. Gas oil is also produced at a loss. Gas oil is not different from fuel oil generically, and gas oil is burned. It, too, is fuel oil. There are certain special markets for parts of the gas oil. For instance, many plants make gasoline out of natural gas by absorption. The absorbent used is gas oil. The term "gas oil" ordinarily is used to designate any oil in the crude that is too heavy to burn in a wick-lamp and not viscous enough to use as a lubricant; and that amounts to a pretty fair proportion of the total amount of the crude. At the outset, it was a very cheap cracker stock. Most cracking processes made use of it, and the spread of cracking processes has reached such proportions today that they are affecting or have affected the price of it. So we look down one step farther to the cheaper product, which is fuel oil. The effect, however, has been very slight up to the present.

CHAIRMAN LUM:—Aside from its use in cracking processes, what is the present market for fuel oil?

MR. LEAMON:—Fuel oil, as the name indicates, is fuel. It is used in competition with coal to a certain extent, but there are a number of furnace uses for fuel that oil will meet and coal will not. In the steel industry, when concentrated heat is needed somewhere beside on the grate bed, a volatile fuel must be used; but the thing that controls the price of fuel oil is the competitive price of coal. Companies pay a premium of approximately 15 or 20 per cent for oil rather than burn coal.

CHAIRMAN LUM:—Will the extension of cracking have the effect of raising the price of fuel oil to the consumer?

[3] M.S.A.E.—Consulting automotive engineer, City of Washington.

MR. LEAMON:—Yes; but another factor that will be much more to the point from that angle is the increased quantity that will be burned. I suppose you have in mind domestic fuel-oil, primarily. That is a very new business which is capable of expansion to enormous proportions. Competition, for the present, will have a greater effect on the price of oils that are supplied for domestic burners than will the cracking process.

A MEMBER:—How does the price by your process compare with the average cost of manufacturing gasoline?

MR. LEAMON:—That is touching on another large subject. When an oil is heated up to its decomposing point, about one-third of it becomes motor fuel, about one-sixteenth of it becomes fixed gas, about one-twenty-fifth of it becomes tar, and the remainder, on the average, would distill about as the original did. Within negligible limits that always occurs, with all processes and with all cracker stocks. The amount of variation is slight. In actual practice, the 35 per cent of conversion was a theoretical and absolute equilibrium figure. Actual processes do not reach that figure. They cannot afford the time.

I have called attention to the fact that the reaction factor is largely a time factor. So, the differences between cracking processes are in the cost of the equipment and the amount of conversion that is obtained. By proper installation, they all can be brought to approximately the same conversion. If one process gives a larger ultimate yield of motor fuel than another, it has therefore an advantage. All processes can be made to give the same ultimate yield but, as the oil is recycled, the cost goes up and, if recycled more than three times, the cost is beyond the economic limit. In other words, if cracker stock is put through the machine and part of it is brought out as motor fuel on each passage through, and if the remainder of the stock is recycled, the economic limit will have been practically reached when the "remainder" has made its third passage through. So, the process yields that are practical are the yields that can be obtained with not to exceed three cycles. A little edge may be developed by certain processes that can increase the yield from those cycles, perhaps.

Vapor-phase yields are always larger than liquid-phase yields, using the total factors all the way through the system, that is, the total conversion from the distillation range of the original cracker stock, including fixed gas, motor fuel and higher-boiling oils. In our case, we have a catalytic process that reduces the time of the reaction as well as controlling the composition of the product.

A MEMBER:—That is a patented feature?

MR. LEAMON:—Yes. Patent applications have been filed but the patents have not been issued. The process is secret.

Reverting to the other condition, namely, the cost of the equipment that is required, all the other systems described employ pressure. At the temperatures employed, very heavy materials are required for the construction of the apparatus; otherwise, it would blow up. A great deal of apparatus has been blown up with the attending fatalities. It is very much worse than when a boiler explodes. Steam and hot water do not burn but the oil does.

In this process we use no pressure; there is no pressure on the reaction element other than the back-pressure caused by the flow of the oil and vapor through the equipment. The construction is light. If something blows up, it releases only vapor; there is no liquid pres-

ent to expand many times and scatter over a large area; consequently, the safety factor is very high, building against high pressures is unnecessary, and standard materials can be used. According to the catalog, it is "out of stock" and we try to buy it in that way. The first cost is pretty low, about 33 to 50 per cent on the dollar, for what the same units could be built for if they were operated at a pressure of 300 lb. per sq. in.

A MEMBER:—Do cracked compounds have a tendency to break up further after cracking?

MR. LEAMON:—No. The product is more stable after it comes through the decomposing zone, so far as the greater part of the constituents present are concerned, but, when coming out of the reaction zone, small quantities of products are always present that are not exactly members of the four families. There are black sheep among the cousins of the families. They are not stable and must be removed. That is the function of treating-processes.

The presence of these black sheep has resulted in a very unjust odium being imposed on one good little family, the Olefins. It was thought at one time, in the early days before chemists began working in oils, that the unsatisfactory constituents in the cracked products were olefins. Now we call them diolefins to differentiate them; there are still better reasons, but that one is sufficient for this discussion. If you found an ore that contained a large quantity of silver and a small amount of copper you would call it a silver ore. If it had more copper than silver you might still call it a silver ore.

The small amounts of so-called diolefins present in cracked products and the very incomplete chemical knowledge of the family resulted in the olefins' getting a black eye. It is commonly considered not to have the antiknock value of the naphthenes. I have not been able to note a difference. I have seen a record of a difference of one-tenth of a compression-ratio ascribed to hexene and cyclohexane but, for all practical purposes, they are the same. The instability of the diolefins may have been partly responsible for the misconception as to the antiknock value of the olefin family.

A MEMBER:—I have seen some cracking tests that show a further breaking up.

MR. LEAMON:—All processes do show it unless those relatively small amounts of diolefins are removed. The thing that really happens is something like two prickly pears' falling together. In the illustration, each carbon atom had four chemical bonds radiating from it and each hydrogen atom only one, and there is a tendency for each of those chemical bonds of the carbon atom to become attached to something beside carbon. It does not evince that tendency radically until after more than two bonds have become attached to other carbon atoms. Sometimes they are not unstable even then. For instance, in benzol, one of the highest antiknock hydrocarbons and fairly stable, all the carbon atoms have three bonds connected with carbon, with a double bond between them, and one other carbon atom. When such a carbon atom has nothing but carbon to associate with, it may combine with another such carbon atom. This combination, forming a molecule of perhaps twice the size of the original, may take on a third, and a still larger molecule compound will be built up, which is called "gum."

It is interesting to note that, although they are present in all cracked products and vary somewhat in quantity, gums have a potential commercial value. A sort of chicle, which forms a wonderful chewing-gum, has been made. By oxidation it can be converted into shellac.

A MEMBER:—It is not oxidized?

MR. LEAMON:—No; not usually. These double bonds will frequently break and take hold of an oxygen molecule, if it is there, but ordinarily in an oil-tank there is not much oxygen. Most of the hydrocarbon molecules are in contact with other molecules of the same kind.

A MEMBER:—After the burning?

MR. LEAMON:—After. If you can get it as far as the combustion-chamber of the engine, it will not cause any trouble. You will find gums on the intake-valves but not on the exhaust-valve. They are subject to oxidation just as are any other oils and their quantity is very small. They are probably more susceptible to oxidation than most of the other heavier oils.

A MEMBER:—Is it not true that a vapor-phase plant requires more area than a liquid-phase plant of the same capacity?

MR. LEAMON:—Quite the reverse in this case. Our 250-bbl. unit occupies a space 30 x 105 ft., including the cracker house at the end and the roadway on both sides. The area for a given capacity is partly a matter of design and partly a matter of time factor.

Cracking as much material by the vapor-phase method in a reacting chamber without a catalyst to speed up the process would require a pretty good-sized reaction element, but the catalyst has reduced the time of reaction to very much less than the liquid-phase reaction, enough to compensate almost entirely for the expansion of volume necessary in the vapor-phase. The original vapor-phase plants were rather bulky, even though they were operated at a pressure of from 100 to 150 lb. per sq. in.

A MEMBER:—The catalyst takes care of that?

MR. LEAMON:—Yes; the catalyst also enables us to make the product free from paraffin. We have practically no paraffin.

A MEMBER:—In the blended type of gasoline, is carbon formed in the same manner as in the average type, but on account of its antiknock features does not result in knocks? Is not the carbon formed as readily as with straight-run gasoline?

MR. LEAMON:—So far as I know, the antiknock feature has nothing to do with the deposition of carbon. If the engine contains carbon and is knocking, part of the clearance space in the top of the cylinders is filled up and the compression-ratio is higher than it should be. If an adequate antiknock fuel is put in, it will not knock. It has been pretty well established that, if an engine that contains carbon is taken out on the road and the throttle is opened wide for a few miles with a lean mixture, the carbon can be eliminated. Whether the carbon is deposited or not is more a function of the mixture than of the fuel.

A MEMBER:—What compression-ratio is used with ordinary gasoline?

MR. LEAMON:—The ordinary automobile engine of today has a compression-ratio of about 4.25. Some have a good deal higher ratio but have volumetric inefficiency to offset it. Stellarene having an end-point of 437 deg. will probably begin to detonate around 5.25 or 5.50. The volatility of the fuel has considerable to do with the compression that it will stand. As has been shown in Fig. 13, the gap between the curves of U. S. Motor Stellarene and Domestic Aviation Stellarene is three or four times as wide as the gap between the dotted curves of the 30 and 40-per cent benzol blends, respectively. The spread between them was 2.8 lb. and the spread between 437 and 390-deg. end-point Stellarene looked as if it might be about three-quarters as much as the step from the paraffin gasoline to the Stellarene curves. This is due to their difference in volatility.

It is very important, in comparing the antidetonation characteristics of fuels, that the volatility be known and that they have practically the same distillation curve. It is not necessary to pay attention to the specific gravity.

A MEMBER:—Straight Stellarene would not knock as readily as ordinary gasoline?

MR. LEAMON:—Not by a difference of several pounds in the compression pressure.

A MEMBER:—With what percentage of benzol would that compare in straight-run gasoline?

MR. LEAMON:—On the average, about 35 per cent. The benzol equivalent of Stellarene is between 30 and 40 per cent in paraffin gasoline; not California gasoline, because California gasoline contains aromatics.

A MEMBER:—All racing-cars use gasoline from California.

MR. LEAMON:—Yes; but I have furnished several of them with Stellarene many times. They require an antiknock fuel, but they do not require unusual antiknock fuel, because the very high speed of rotation makes high volumetric efficiency impossible and 390-deg. end-point Stellarene has been run. One of these cars finished in third or fourth place in the 500-mile race at Indianapolis in 1925 with an 8.5 to 1 compression-ratio and was in good shape at the finish, as such engines go. It was not ready to enter another race, but was not damaged.

A MEMBER:—What is the retail price of Stellarene?

MR. LEAMON:—Up to the present, considering the size of the motor-fuel market, we have produced only a small amount but we have done some experimenting. It has been marketed at three different prices for test purposes, at a 2, a 3 and a 5-cent premium. That is where I derived the data by which I dared to say that 20-per cent premium is about the psychological limit. Five cents is too much. Three cents was satisfactory. We had to put some sales effort behind it to move it at 5 cents. At 3 cents it sold itself, and, of course, at 2 cents, it sold itself also.

A MEMBER:—Three cents was what percentage of the cost?

MR. LEAMON:—Three cents a gallon premium.

A MEMBER:—Was that in the South?

MR. LEAMON:—No; in Ohio I think that 4 cents would probably be about right. We have not yet had a chance to try 4 cents, but I think it is worth the sales effort to put it out at 5 cents and get the extra cent.

A MEMBER:—In the market where Stellarene was sold at various prices, what was the premium asked for benzol blends?

MR. LEAMON:—We did not have any direct competition with benzol blends at that time. Three years ago in Ohio, we did not have benzol blends. We hardly ever heard of them. One of the firms operated from Washington, Pa., which is about 30 miles south west of Pittsburgh. In Pittsburgh, they were marketing benzol blends at the time, but we could not call that competitive comparison.

The benzol or benzol blend being marketed at that time gave trouble, because they had not yet learned to refine benzol. Benzol had to be refined as well as gasoline and the product sold at that time was not very satisfactory and would not have been competitive even in the same town.

A MEMBER:—Did you say Stellarene would give approximately 20 per cent more mileage?

MR. LEAMON:—Yes. We have run road tests on hundreds of automobiles. There is no theoretical reason that more mileage should be obtained with antiknock fuel, unless the engine in which it is used is overcompressed and is causing detonation losses to occur with ordinary fuel. You do not get more mileage with one antiknock fuel than with another; on the average it is about the same, provided the antiknock fuel has sufficient antiknock value. Twenty per cent is the average figure; the number of tests that have shown between 19 and 21 per cent is remarkable, that is, tests not only on road runs of 100 or 500 miles but on tests of truck fleets over periods of months. They all come back to the same thing.

A MEMBER:—What happens in cold weather? What percentage of benzol is used?

MR. LEAMON:—The percentage of benzol required to affect the detonation in the fuel depends upon the gasoline with which it is blended.

A MEMBER:—The better the grade of gasoline, the smaller the amount of benzol?

MR. LEAMON:—The quality of the fuel governs the amount of benzol. No method has yet been devised so that it can be done as readily as we should like. Benzol blends like Betholene and Amico both are close to 50 per cent and are equivalent to the ordinary 50-per cent-benzol blend. Benzol blends with California gasoline usually have a low end-point.

TAXATION

THE real tax problem in the United States today will be found in the county seats, city halls and State capitals, rather than in the City of Washington. Federal taxes have now been reduced to a point where, either from the standpoint of rates or of total volume, they no longer constitute an excessive burden for a nation as rich as ours. Raising $2,500,000,000 in internal revenue taxes by the Federal Government is not a problem of any magnitude, but in 1924 our States and localities raised by taxation $4,812,000,000. From 1919 to 1924 Federal taxes were reduced by $1,974,000,000 per year, while those levied by the States and the local taxing bodies increased $1,847,000,000 and unquestionably are continuing to increase at a rate of probably not less than 10 per cent per year.

Over 90 per cent of the $3,478,000,000 of local taxes rests on real property. In New York City, it has been estimated that anywhere from 2 to 3 months' rent is not rent at all but taxes, a condition that is particularly burdensome where the supply of cheap housing facilities is limited and where a tenement house problem exists. In the rural districts, taxes constitute one of the principal contributing factors to the high cost of production and consequently low profits. In New York, property taxes are consuming from 30 to 50 per cent of net income from property in the prosperous agricultural sections of the State. What is true in New York is unquestionably true throughout the Country. It constitutes the real tax problem in the United States.—Ogden L. Mills in *Trade Winds* (Union Trust Co. of Cleveland).

Some Factors That Affect the Frictional Properties of Automobile Brake-Linings[1]

By H. H. Allen[2]

Annual Meeting Paper

Illustrated with Drawings and Photographs

ABSTRACT

VARIATION of the retarding forces on the brakes of automotive vehicles for a given pedal position or a given pedal pressure is frequently due to the ordinary wear of the brake-linings or other parts and may readily be prevented by periodic inspection and adjustment. Sometimes, however, sudden and serious reduction in retarding ability occurs when the brakes have been applied for comparatively long periods with short cooling-intervals or when the brakes have been wetted or oil has reached the linings. Laboratory tests of braking materials have shown that a marked increase in temperature will generally result in a reduction in the coefficient of friction of asbestos textile brake-lining materials and that oil and water have a similar effect of a temporary character. In the present paper laboratory tests have been paralleled by a series of experiments undertaken with a view to securing data on the working conditions of the brakes of a passenger-car in actual service, particular attention being given to the nature and magnitude of the causes that lead to variations of braking ability. Three series of tests were made: (*a*) with the brake-linings dry, (*b*) after the linings had been soaked with water and (*c*) after the linings had been dried and had been treated with liberal potions of oil.

A detailed description of the testing apparatus is given, two features of which were the thermocouple set-up, consisting of eight copper-constantan couples, one in each brake-shoe, and a decelerometer for measuring the deceleration obtained in feet per second per second as the car was brought to rest. The results of the tests are explained and are depicted graphically in a series of curves.

The conclusions reached are that (*a*) with linings considered representative of those in general use in automotive vehicles, the apparent coefficient of friction drops with rise of temperature; (*b*) the drop in the apparent coefficient of friction with rise of temperature may be largely, if not wholly, due to the influence of the temperature rise upon the saturant; (*c*) the drop in the apparent coefficient of friction with rise of temperature seems to be less for hard, dense linings than for others; (*d*) no typical difference in performance with change of temperature was apparent between the woven asphaltic-saturated linings and the rubberized, or vulcanized, folded and stitched linings; (*e*) the tendency of the apparent coefficent of friction to rise rapidly seems to be greater for the rubberized or vulcanized linings after being water-soaked than for the solid woven asphalt-saturated linings; neither increase nor decrease of the coefficient of friction with water-soaking was apparent; and (*f*) the coefficient of friction of all linings tends to become less after they have been oil-soaked, and their tendency to recuperate from such a condition is not rapid.

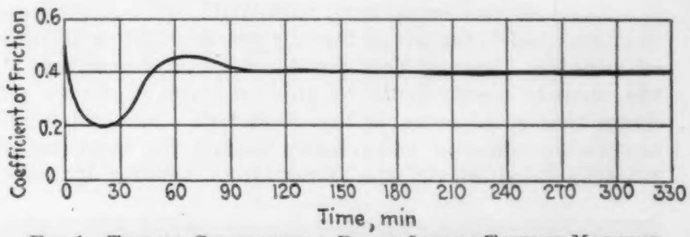


Fig. 1—Typical Curve from a Brake-Lining Testing-Machine

The Coefficients of Friction Have Been Plotted as Ordinates and the Time in Minutes as Abscissas. The Coefficient Drops during the First 20 or 25 Min. of Running, during Which a Large Percentage of the Volatile Constituents of the Saturant Is Driven Off, Then Rises and Finally Assumes a Constant Value, Even Though the Temperature Remains Substantially the Same

THE brakes of automotive vehicles frequently show variations in performance that are evidenced by variation of the retarding force for a given pedal position or pedal pressure. When such change is due to ordinary wear of the brake-linings, or other parts, the change is gradual, and serious results may readily be prevented by periodic inspection and adjustment.

Even properly adjusted brakes frequently show sudden and serious reduction in retarding ability, however. This may occur during or after long-continued application, as in descending long grades when the brakes are applied frequently for comparatively long periods and the cooling-intervals are short. Similar changes may be noticed when the brakes have been wetted or when oil has reached the linings.

Although the sudden decrease in brake effectiveness is usually only temporary, its consequences may prove disastrous, and an understanding of the nature and causes of such decrease should be of value to the engineer in brake design and may assist the operator to forestall possible serious results.

Laboratory tests of brake-lining materials have already yielded much information in this connection. Such tests show that a marked increase in temperature will generally result in a reduction of the coefficient of friction of the asbestos textile brake-lining materials almost universally used in automotive vehicles.

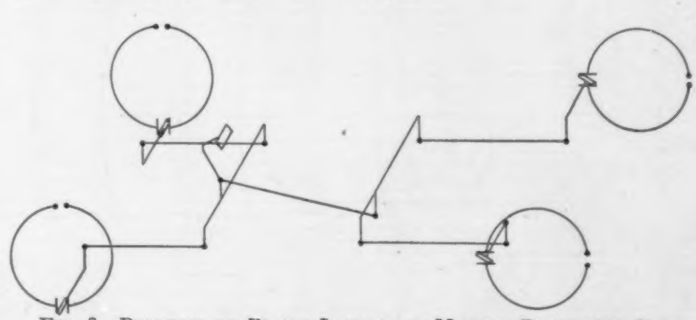


Fig. 2—Diagram of Brake Layout of Modern Passenger-Car Equipped with Four-Wheel Brakes

This Layout Was Used Throughout the Tests. The Brakes Were Pivoted at One End and Were Operated by a Cam

[1] Published by permission of the Director of the Bureau of Standards.

[2] Assistant mechanical engineer, Bureau of Standards, City of Washington.

Fig. 1 is a typical curve from a brake-lining testing-machine, where the coefficients of friction have been plotted as ordinates and time in minutes as abscissas. It can readily be seen how rapid the drop in the coefficient of friction is over a period of about 15 or 20 min., how rapid the rise is again after that period, and how uniform the coefficient remains after the initial period of the run has been completed.

Fig. 1 shows a pronounced decrease in the coefficient of friction and is typical of the results obtained under test conditions that cause a rapid temperature-increase of several hundred degrees. This curve was obtained in connection with the use of a brake-lining testing-machine developed at the Bureau of Standards. It will be noted that the coefficient drops during the first 20 or 25 min. of running. During this period, a large percentage of the volatile constituents of the saturant is driven off. From this point onward, the coefficient rises and finally assumes a constant value, even though the temperature remains substantially the same. With smaller increases

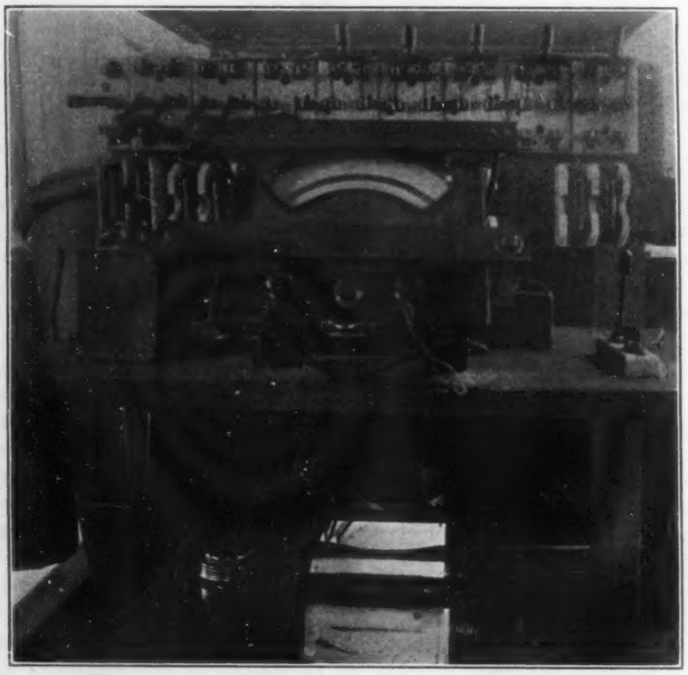


Fig. 3—Thermocouple Set-Up Used in Testing Brake-Linings
The Set-Up Consists of Eight Copper-Constantan Couples, One in Each Brake-Shoe, So Connected That Either the Average Temperature of All the Couples or the Temperature of Any One Couple May Be Read. The Couples, in Conjunction with a Milliammeter, Were Calibrated against a Centigrade Thermometer

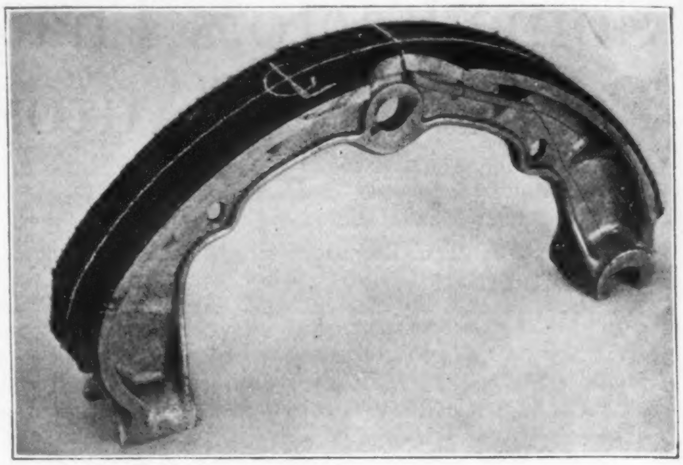


Fig. 4—Photograph of Cutaway Brake-Shoe Assembly
The Hot Junction of Each Thermocouple Was Embedded in the Lining. The Cold Junctions Were Inserted into Mercury at the Bottom of Small Glass Tubes and These Were Grouped in an Ice Bath Near the Switchboard and the Milliammeter

in temperature, the drop in the coefficient of friction is found to be less pronounced. Laboratory tests have shown repeatedly that both water and oil may cause a marked lowering of the coefficient of friction and, in addition, have demonstrated the more or less temporary character of such effects.

These laboratory tests have been paralleled by a series of experiments undertaken with a view to securing data on the working conditions and the operation of brakes on a passenger-car in actual service. In this connection, particular attention was given to the nature and magnitude of the causes that lead to such variations of braking ability as have been mentioned. The object of this paper is to present the results obtained.

Method of Testing

A modern passenger-car equipped with four-wheel brakes, shown diagrammatically in Fig. 2, was used throughout the tests. The brakes were pivoted at one end and operated by a cam. The following method was employed. The car was run for a distance of approximately ¼ mile at constant speed, with the brake-pedal depressed to a fixed stop, and the resulting temperature was measured by thermocouples set in the linings of each brake-shoe. Without releasing the brakes, the clutch was disengaged, the deceleration was recorded by a recording decelerometer, and the pedal pressure was obtained simultaneously by a gage attached to the brake-

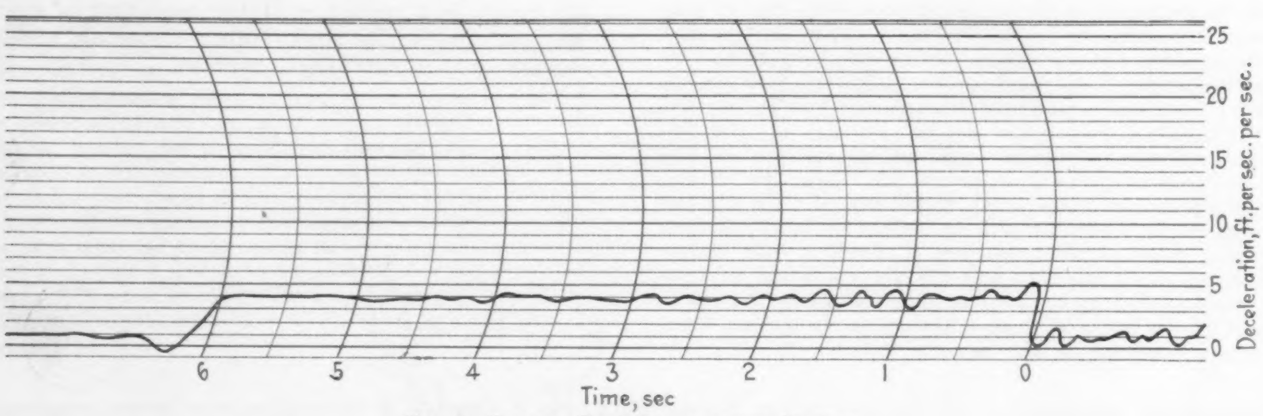


Fig. 5—Typical Time-Deceleration Record
The Time, in Seconds, Starts from the Right-Hand Side and Moves from Right to Left. The Curve Shows the Deceleration in Feet per Second per Second as the Car Is Brought to Rest. The Car Is Traveling at Uniform Speed at the Right and Comes to Rest at the Left

pedal. This was repeated at each of several brake-pedal pressures up to the maximum at which the engine would carry the load.

In another series of tests, the linings were thoroughly drenched with water. Successive decelerations were then made, the same measurements being taken as previously indicated. All these decelerations were made at one position of the brake-pedal and, consequently, at substantially equal pressures.

After the linings and drums had become thoroughly dry, they were treated with liberal potions of oil, and another series of successive decelerations was made in the same manner as described above.

Description of Apparatus Used

The apparatus alluded to above will now be described in greater detail. The thermocouple set-up consists of eight copper-constantan couples, one in each brake-shoe. These are so connected that either the average temperature of all the couples, or the temperature of any one couple, may be read. A photograph of the set-up is shown

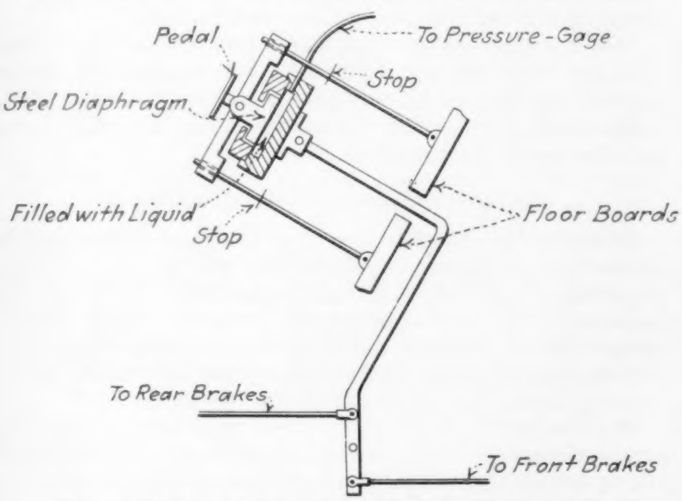


Fig. 6—Method of Operating the Pedal-Pressure Gage

The Instrument Consists of a Thin Steel Diaphragm Pressed Upon by a Block on Which Is Mounted the Brake-Pedal Pad. The Force upon the Diaphragm Is Transmitted to the Liquid in a Closed Hydraulic System and Is Indicated by a Pressure-Gage. The Pressures Were Not Read Directly in Pounds But the Instrument Was Calibrated So That Pounds Could Be Interpolated from the Readings of the Gage. Stops Were Provided So That the Pedal Position and Hence the Pressure Might Be Maintained Substantially Constant During the Time the Brake Was Applied

in Fig. 3. The couples, in conjunction with a micro-ammeter, were calibrated against a centigrade thermometer. The hot junction of each couple was embedded in the lining, as is shown in the photograph of a cutaway brake-shoe assembly, Fig. 4. The cold junctions were inserted into mercury at the bottom of small glass tubes and these were grouped in an ice bath near the switchboard and the milliammeter. The center line of the brake, shown at the center of the shoe, is half-way between these two points. The thermocouple was inserted at that point, half-way through the lining. The couples were protected by rubber connections and the junctions were brought out to the main switchboard.

The decelerometer used throughout this work is the one that was developed by the Bureau of Standards and has previously been described[3]. Suffice it to say that this instrument is one by which records of deceleration may be obtained in feet per second per second. A typical time-deceleration record is shown in Fig. 5.

The time, in seconds, starts from the right-hand side

[3] See The Journal, December 1923, p. 499.

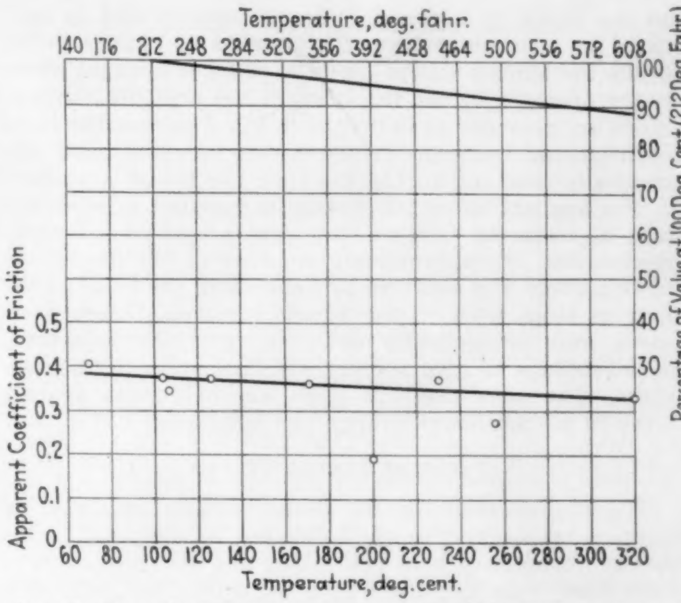


Fig. 7—Curve Showing the Influence of Temperature on the Apparent Coefficient of Friction of Lining No. 153

The Curves Reproduced in This Chart and in Fig. 8 Show the Variation of the Coefficient in Percentage of Its Value at 100 Deg. Cent. (212 Deg. Fahr.) The Arbitrary Selection of 100 Deg. Cent. as Unity Was Made Because It Approximated the Lowest Temperature for Which Data Were Secured for All the Linings and Serves as the Most Convenient Criterion for Intercomparison of Performance of the Different Linings

and moves from right to left. The curve shows the deceleration in feet per second per second, as the car is brought to rest. The car is traveling at uniform speed at the right and comes to rest at the left.

Pedal pressures were obtained with a pedal-pressure gage designed for the purpose. The pressures were not read directly in pounds, but the instrument was calibrated so that pounds could be interpolated from the readings of the gage.

This instrument consists of a thin steel diaphragm pressed upon by a block on which is mounted the brake-pedal pad. The force upon the diaphragm is transmitted

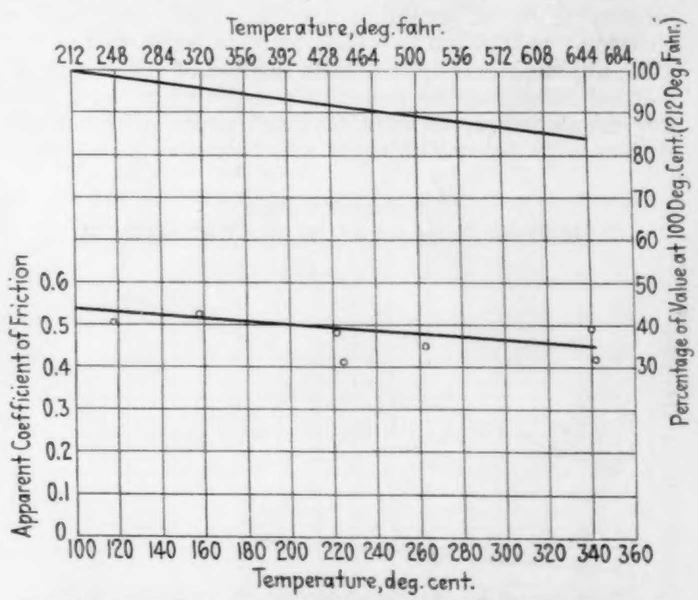


Fig. 8—Curve Showing the Influence of Temperature on the Apparent Coefficient of Friction of Lining No. 161

The Curves Reproduced in This Chart and in Fig. 7 Show That, Starting with the Value at 100 Deg. Cent. as Unity, the Apparent Coefficient of Friction Decreases with Rising Temperature at a Rate Varying from the Minimum of About 5 Per Cent per 100 Deg. Cent. to the Maximum of About 50 Per Cent per 100 Deg. Cent. The Average Rate of Decrease for Linings Nos. 152, 153, 154, and 161 Is About 19.5 Per Cent per 100 Deg. Cent.

to the liquid in a closed hydraulic system and is indicated by a pressure-gage. This makes no change in the brake mechanism except that the pedal is brought closer to the operator by the thickness of the pressure element. Stops are provided as indicated in Fig 6 so that the pedal position, and hence the pressure, may be maintained substantially constant during the time the brake is applied.

The application of the brakes is resisted to some extent by retractor springs that form a part of the brake mechanism. An approximate measure of the magnitude of this effect was obtained by depressing the pedal to the various stops with all the wheels removed. These pressures were subsequently deducted from the total-pressure readings to give the net effective brake-pedal pressures. The pedal-pressure gage was calibrated against force in pounds exerted upon the pedal-pad.

Types of Lining Tested

The linings used for the different tests consisted of samples representative of those used in every-day practice and comprised both the folded and stitched, rubber-vulcanized type and the woven asphaltic-base saturated type. These two types of lining included linings representing various degrees of hardness, density and the like so that differences of performance due to these various characteristics might be investigated.

The true coefficient of friction of the brake-lining is given by the ratio of the total retarding-force at the brake-drums to the total normal, or radial, force acting against the drums. Since the total normal force is not readily measurable, and since this force is nearly proportional to the pedal pressure, which is readily measurable, it is assumed that the normal force is proportional to the pedal pressure. Although this assumption is in error by greater or lesser amounts, influenced by the coefficient of friction and other variables, we believe, nevertheless, that the deviation of the normal force thus computed from the actual normal force is not sufficiently erratic to change the general trend of the results. The ratio of the total retarding-force at the brake-drums to the total normal-force, thus computed, against the drums, is hereafter called the "apparent coefficient of friction." Attention is called at this point to the fact that, although this approximate method of computing the normal force may be reasonably satisfactory as a basis for comparing coefficients obtained under different conditions with a given lining, it may introduce considerable

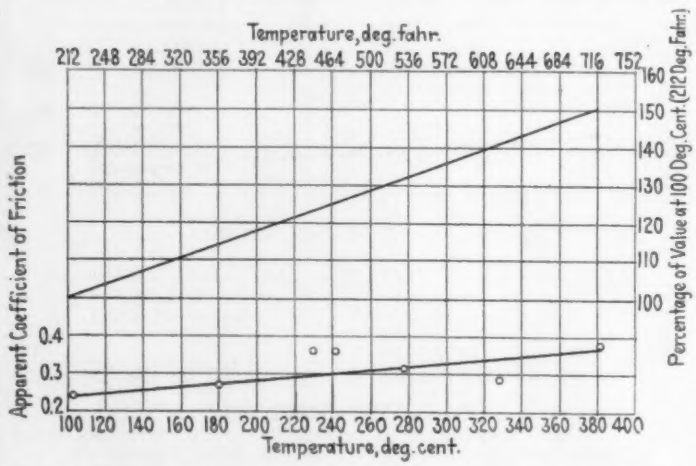


FIG. 9—CURVE SHOWING THE INFLUENCE OF TEMPERATURE ON THE APPARENT COEFFICIENT OF FRICTION OF LINING NO. 224

This Is the Only Lining Tested That Showed an Increase in the Value with Rise of Temperature. It Was an Asbestos Textile Lining That Was Closely Similar to One of the Others But Without Any Saturant

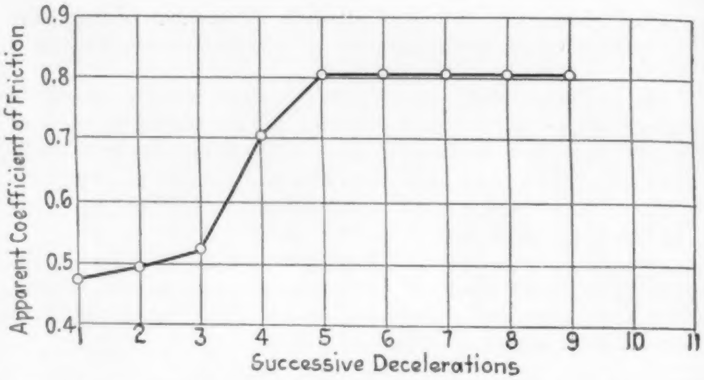


FIG. 10—INFLUENCE OF WATER ON THE APPARENT COEFFICIENT OF FRICTION OF LINING NO. 154

This and the Curves Shown in the Two Following Charts Show a General Increase in the Apparent Coefficient of Friction with Successive Decelerations, Although the Rate of Increase Varies to Some Extent

uncertainty in an intercomparison of different linings. There is, therefore, no absolute reference datum as between the several tests on different linings.

For the purpose of computing the normal, or radial, pressure from the pedal pressure, the mechanical advantage of the braking system was taken as the ratio of the brake-pedal travel to the corresponding average movement between the brake-shoes.

Average Temperatures Plotted

Attention is called to the fact that the temperatures plotted in the curves represent the average of the temperatures of the lining at each brake-shoe. Although the pressures and the temperatures at each of the several shoes were not exactly the same, nevertheless they were substantially so, making the assumption of similar conditions at each shoe and brake not a violent one.

The results obtained in the first tests seemed to indicate a tendency of the coefficient to drop not only with rising temperature but also with increasing pressure at supposedly constant temperature. As this behavior seemed abnormal, further runs were made after first inserting additional thermocouples at the rubbing surface of the lining and the drum and on the under side of the lining. When the readings of the three couples had been plotted against time, it was readily seen that there was a considerable temperature-gradient through the lining and that a longer time was required to reach temperature equilibrium. The running-time was consequently increased to 5 min. at each pressure, and the time between decelerations was made as short as practicable. All tests were made at 15 m.p.h.

Another change, namely, the removal from the brake mechanism of all springs not absolutely essential to its operation, was made, to cut down the brake-pedal force necessary to flex them. The purpose was to make the brake-pedal pressure correction as small as possible. At the lighter pressures, it had been a considerable proportion of the total brake-pedal pressure.

Grouping of Linings

For more convenient reference, the curves showing apparent coefficient of friction versus temperature will be grouped together for all the linings, and those showing results of successive decelerations after application of water and of oil will be shown later, similarly grouped.

For convenience, the linings are grouped according to their various structural and physical characteristics: linings Nos. 152, 153, 156, 160, 161, and 224 are of the

woven type of asbestos material, and linings Nos. 154 and 157 are of the folded and stitched vulcanized or "rubberized" type. Linings Nos. 153, 156 and 161 differ from the remainder in being much harder and less pliable. Lining No. 224 is woven of exactly the same material as lining No. 161 but is not treated with saturant after coming from the loom.

Influence of Temperature Changes

In Figs. 7 and 8, which show the influence of temperature upon the apparent coefficient of friction, curves are presented that show the variation of this coefficient in percentages of its value at 100 deg. cent. (212 deg. fahr.). The arbitrary selection of 100 deg. cent. as unity was made because it approximated the lowest temperature for which data were secured for all the linings. This value at 100 deg. cent. would then serve also as the most convenient criterion for intercomparison of performance for the different linings, within the limitations already set forth.

The curves show that, starting with the value at 100 deg. cent. as unity, the apparent coefficient of friction decreases with rising temperature at a rate varying from the minimum of about 5 per cent per 100 deg. cent. to the maximum of about 50 per cent per 100 deg. cent. The average rate of decrease for linings Nos. 152, 153, 154, and 161 is about 19.5 per cent per 100 deg. cent.

Only one lining tested showed an increase in the value of the apparent coefficient of friction with rise in temperature, see Fig. 9. This was lining No. 224, an asbestos textile lining closely similar to one of the others but without any saturant. In this lining we have an increase of about 50 per cent over the value of the apparent coefficient of friction at 100 deg. cent. The marked difference in behavior of lining No. 224 from that of the saturated linings seems to show that decreases in the coefficient of friction of the finished linings may be largely, if not altogether, due to the influence of the temperature rise of the saturant. It may be stated that laboratory tests also show less drop in the coefficient of friction for untreated than for treated linings.

The need of a saturant is indicated by the fact that the untreated lining showed greater changes in character and appearance than any of the others. It had become blackened, its cotton content was charred at the surface and was considerably frayed out, whereas the other linings showed no evidence of the high temperatures attained, only the usual evidences of wear from hard usage being noticeable.

On the basis of these tests, no conclusion can be drawn as to a relation between the change in the apparent coefficient of friction with temperature and the mode of manufacture of the lining. Greater differences in the percentage change of the apparent coefficient of friction with temperature seem to occur between different linings having similar methods of manufacture than between the different types.

On the other hand, the two lowest drops in the apparent coefficient of friction and also the least percentage-drops are shown by the harder and less pliable linings, No. 153 (Fig. 7) and No. 161 (Fig. 8).

Influence of Water and Oil

In Figs. 10 to 12, inclusive, the apparent coefficient of friction of different linings is plotted against the number of successive decelerations made, after the linings had been thoroughly drenched with water. All these decelerations were made with approximately the same pedal-pressure. These curves show a general increase in the apparent coefficient of friction with successive decelerations, although the rate of this increase varies to some extent.

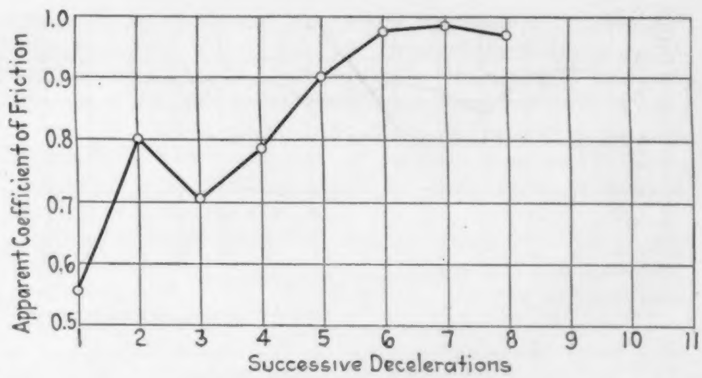


Fig. 11—Influence of Water on the Apparent Coefficient of Friction of Lining No. 157

The Principal Difference Noticeable in the Curves Shown in Figs. 10, 11 and 12 Is the More Rapid Rise of the Coefficient of Friction of the So-Called "Rubberized" Linings As a Result of the First Few Decelerations. This Type of Lining Is Shown in Figs. 10 and 11

The principal difference noticeable is the more rapid rise of the apparent coefficient of friction of the so-called "rubberized" linings as a result of the first few decelerations. The performance of this type of lining is shown in Figs. 10 and 11.

Temperature measurements were made at the end of each deceleration during the "water-wetted" runs; but these temperatures have been plotted only for lining No. 161, as shown in Fig. 12. The values shown are only approximations of the temperatures at the rubbing surfaces, as the time consumed was insufficient for attainment of temperature equilibrium in any case.

Fig. 13 shows the apparent coefficient of friction during a number of successive decelerations made with lining No. 154 after it had been thoroughly soaked with a heavy engine-oil. The series of tests with oil-soaked linings shows much smaller differences, as between linings, than was the case with the water-soaked linings. Regardless of the type or character of the lining, its ability to recover its previous frictional properties in from 10 to 16 decelerations was noticeable chiefly by its absence. In other words, oil seems to be the general leveler of brake-lining materials, so far as their frictional qualities are concerned, at least for a short period after the oil has been applied.

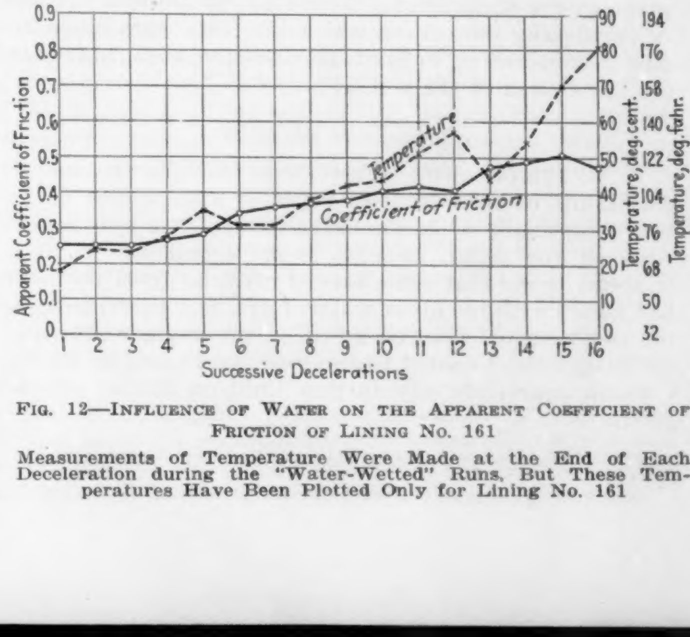


Fig. 12—Influence of Water on the Apparent Coefficient of Friction of Lining No. 161

Measurements of Temperature Were Made at the End of Each Deceleration during the "Water-Wetted" Runs, But These Temperatures Have Been Plotted Only for Lining No. 161

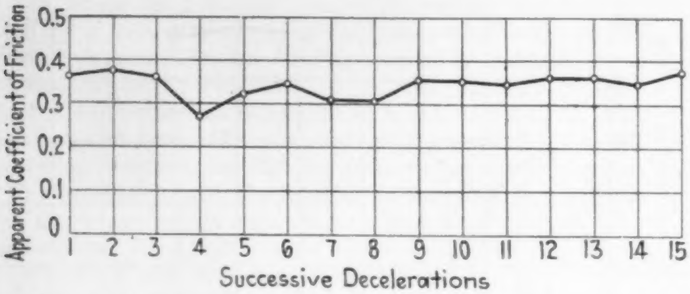


FIG. 13—INFLUENCE OF OIL ON THE APPARENT COEFFICIENT OF FRICTION OF LINING NO. 154

This Curve Shows the Variation of the Apparent Coefficient of Friction during a Number of Successive Decelerations Made After the Lining Had Been Thoroughly Soaked with a Heavy Engine-Oil. The Series of Tests with Oil-Soaked Linings Shows Much Smaller Differences, as between Linings, Than Was the Case with the Water-Soaked Linings

CONCLUSIONS

From analysis of the foregoing it seems that

(1) With linings considered representative of those in general use in automotive vehicles, the apparent coefficient of friction drops with rise of temperature

(2) Since the only lining tested that failed to comply with the foregoing statement was one untreated with any saturant or binder, it seems likely that the drop in the apparent coefficient of friction with rise of temperature may be largely, if not wholly, due to the influence of the temperature rise upon the saturant

(3) The drop in the apparent coefficient of friction with rise of temperature seems to be less for hard dense linings than for the others

(4) No typical difference in variation of performance with temperature seems to exist as between the woven asphaltic-saturated linings and the rubberized or vulcanized, folded and stitched linings

(5) There seems to be a greater tendency for the apparent coefficient of friction to rise rapidly for the rubberized or vulcanized linings after being water-soaked than for the solid woven asphalt-saturated linings There seems to be no general indication whether water-soaking will either increase or decrease the coefficient of friction

(6) There seems to be a marked tendency for the coefficient of friction of all linings to become less after they have been oil-soaked The tendency to recuperate from such a condition is not rapid

ACKNOWLEDGMENT

I wish to express my appreciation of the assistance given by the Rickenbacker Motor Co., of Detroit, not only in furnishing the car on which the tests were made but also in cooperating to the fullest possible extent throughout the course of the work.

THE DISCUSSION

W. R. STRICKLAND[4]:—These tests showing the various coefficients of friction after repeated applications of the brake, especially when the brake-linings have been soaked with oil and water, indicate to me a tendency to grab. It seems to me that some eroded material from the band has joined with the oil or water, forming a paste, to make the coefficient of friction go up in the manner shown in the diagrams. I cannot understand the results as shown. I would appreciate any further light on it that can be given.

H. H. ALLEN:—The only explanation I can give for the apparent coefficient of friction's rising after a number of successive decelerations is that, particularly with water, the water was driven off and a more normal surface was presented as a rubbing surface for the drum than had previously been the case.

MR. STRICKLAND:—No point was shown on the curve at which the water or oil was actually driven off and the material returned to its normal condition.

MR. ALLEN:—Some of the curves were repeated a sufficient number of decelerations so that the materials did come back to their normal conditions. If the number of decelerations had been increased still further, however, the results would have been vitiated, inasmuch as we should have been getting temperature effects that we should have been unable to measure, and we should have been running into the same difficulty that we should run into if we should disregard the effect of temperature entirely, which is the most important thing that we are trying to find out.

CHAIRMAN CARL BREER[5]:—How true will the tests check when they are repeated many times? For instance, in wetting a brake, is the result purely a matter of repetition?

MR. ALLEN:—If the pressures are approximately the same, the results repeat very closely.

MR. STRICKLAND:—One conclusion, it seems to me, that has been left out is that by feeding water or oil on the brake-band, and then partly drying the band by application of the brakes, the coefficient of friction would be increased and the braking would be much improved, though touchy.

MR. ALLEN:—The contrary conclusion could be drawn from some of the curves, because it was shown in every case that the coefficient of friction between the rubbing surfaces was less at the beginning, or at the water-soaked portion of the runs, than after the surfaces had become dry; that is, a rise in the coefficient of friction seemed to occur, after the water had been driven out, that was different from the condition in which some film or boundary lubrication was present.

CHAIRMAN BREER:—I think the curve will show a higher coefficient of friction when the brake surface is dry than when it is wet.

MR. STRICKLAND:—I think it will; but a feature of brake operation that most of us are familiar with is grabbing; and it is a very undesirable condition. To prevent that and get back to a uniform basis, the bands must be cleaned or burned off and the sticky substance that is formed by the mixture of oil and the material that comes from the fabric must be eliminated.

MR. ALLEN:—Some of the runs were made after the linings had been wet. Repeated tests of the temperature during the runs were made and no such deleterious effect as you mention was noticed. In every instance, the test was begun with clean new linings.

The sticky gummy substance or paste that you mention would probably be present only after long and hard usage. A great many other results could be interpreted in addition to those which we attempted to interpret, but we have attempted to record the things that we found.

A MEMBER:—If that is the case, one lining that you have shown in the initial test to determine the coefficient of friction did not show a coefficient of friction at any point above 0.5. On the water test, it starts with an initial coefficient of friction of about 0.2 and builds up to around 0.9. Suppose that lining were continued on test,

[4] M.S.A.E.—Assistant chief engineer, Cadillac Motor Car Co., Detroit.

[5] M.S.A.E.—Executive engineer, Chrysler Corporation, Detroit.

would it drop to an average of, say, 0.5 or would it continue up to 0.9?

MR. ALLEN:—Unless the same brake-lining was compared in both instances, it would be pretty hard to say whether an effect of grabbing was approaching, inasmuch as we made an approximation of the coefficient of friction that was computed. It was not the actual coefficient, but merely the apparent coefficient; and, as has been pointed out in the paper, intercomparisons of different linings would be rather difficult if the apparent and not the actual coefficient of friction were used.

The reason the actual coefficient of friction was not measured was that it would require a rather complicated series of computations, in the first place, to determine what the coefficient of friction was for a given temperature; and at every change in temperature that same computation would have to be repeated. For that reason, intercomparison of different linings would be rather difficult, so that a coefficient of friction of 0.9 for one lining cannot be compared with 0.5 for another.

A MEMBER:—I had reference to the same lining, shown under different test-conditions.

MR. ALLEN:—It might be possible that such a grabbing effect as Mr. Strickland points out does actually take place.

K. J. HOWELL[6]:—Were the rivets flush or were they countersunk?

MR. ALLEN:—They were countersunk.

L. C. HUCK[7]:—Has anything been done toward finding out how much the difference in expansion of the drum and of the shoe affects the results? In other words, if we plot an apparent coefficient of friction, basing it on the pedal pressure, which is a direct function of the normal pressure, thus changing the distribution of pressure on the liner, which happens if the brake-drum should expand more than the shoe, it seems to me the result would be affected considerably. If, for instance, we should take the brake-drum, the shoe and the whole mechanism and bring it up to a certain temperature by outside means, not by heating it by dragging the brakes, would not the results be different?

MR. ALLEN:—Nothing has been done to check that. An enormous number of variables are present, which, in the nature of the experiment, could not be accurately measured or even computed. The results are comparative for the same lining at the same temperature, and for different linings at the same temperature, in that they show the trend; they are not absolute values. Your point is well taken. If we were to measure the actual coefficient of friction, all the variables would have to be taken into consideration. On the other hand, in the case of internal-expanding brakes, if the drum expanded, the pressure naturally would be less, so that would be partly compensated for.

T. P. CHASE[8]:—Mr. Allen mentioned soaking the linings in oil. Will he state the method and how long he soaked them?

MR. ALLEN:—We had a rather unique method of soaking the linings with water. In Rock Creek Park is a creek having a number of fords through which the car could be run. We took advantage of the topography of the country and ran the car through these fords. To be absolutely certain that the linings had become thoroughly drenched with water, we ran through these fords backward from 12 to 15 times until the linings and the car, and sometimes the passengers, were drenched with water.

MR. CHASE:—I asked that question because it seemed to me that the coefficient of friction was rather high compared with that obtained in some tests of soaked linings that we have made.

MR. ALLEN:—The coefficient of friction in this case is not the actual coefficient of friction but the apparent coefficient of friction, as computed from the assumptions stated in the paper.

J. W. SAFFOLD[9]:—Has Mr. Allen made any tests with so-called waterproof brake-linings?

MR. ALLEN:—I do not know what claims were made for the linings we used, but, as stated previously, the linings were of different types, fairly representative of the different kinds of brake-lining obtainable on the market.

MR. SAFFOLD:—The object of the question was to determine whether, if repeated tests were made with the same linings, the so-called waterproofing showed any deterioration.

MR. ALLEN:—Particular attention was not paid to the condition of the linings, except to observe more or less casually in each case whether the linings had deteriorated badly. In every case, except that already mentioned in which the lining was unsaturated, the lining appeared perfectly normal after all tests had been made, except that, after the linings had been soaked with oil, they presented, naturally, a somewhat different appearance from that of normal lining. But so far as wear and other conditions are concerned, nothing extraordinary was present, so far as could be seen upon casual observation of all the linings tested.

CHAIRMAN BREER:—Have you found any linings that have practically the same coefficient of friction wet or dry, that is, relatively speaking?

MR. ALLEN:—Relatively speaking, the hardest linings are likely to show the least difference in their characteristic apparent coefficient of friction, when dry, wet, water-soaked, or oil-soaked.

MR. STRICKLAND:—Did you test the linings for water absorption?

MR. ALLEN:—No such tests were made in this instance.

A. J. SCAIFE:—Have tests been made with metal-to-metal brake-shoes to show the effect of water or oil?

MR. ALLEN:—No tests, other than those indicated, have been used in this series of experiments. The materials in every case were asbestos textile materials.

MR. HOWELL:—Was any provision made for keeping the lining clean while being soaked? Was there any chance of mud and dirt getting into the brake-drums?

MR. ALLEN:—If any mud or dirt was in them, it would certainly wash out while the drenching process was going on. All the tests that were run after the linings had been drenched were run on asphalt-coated roads, and the fords were brick-paved, so that no mud or foreign material was likely to be encountered. If any was encountered, the tendency was for it to wash off rather than to accumulate.

MR. STRICKLAND:—Was the water clear?

MR. ALLEN:—Sometimes it was clear and sometimes not.

MR. STRICKLAND:—I have seen tests in which the brakes acted somewhat differently when clear water was used than they did when run through the ordinary water encountered on the roads, even rain water.

[6] Jun. S.A.E.—Laboratory engineer, Studebaker Corporation of America, Detroit.

[7] M.S.A.E.—President, Huck Axle Corporation, Chicago.

[8] M.S.A.E.—Research engineer, General Motors Corporation. Research Laboratories, Detroit.

[9] A.S.A.E.—President and sales manager, Devices Development Co., Cleveland.

[10] M.S.A.E.—Field engineer, White Motor Co., Cleveland.

MR. ALLEN:—Although no particular checks were made, the fact remains that the water is the clearest when it is the lowest; when it is the highest, cars are not allowed to run through the fords.

MR. HUCK:—Contrary to the data generally available, I have noticed that a slight amount of moisture seems to raise the coefficient of friction considerably. Did Mr. Allen notice anything like that in his tests?

MR. ALLEN:—No such experiments were carried on. It would be exceedingly difficult to have a slight amount of moisture on one lining and an equal amount on the next. A procedure was gone through that could be repeated several times and that would yield approximately the same results. Incidentally, in view of the oft-asserted effect of a rise in decelerating ability of linings after they have been wetted, I have made a number of experiments and, in every case in which the driver said that he noticed that the car braked better after the lining had been wet, he always said that the car behaved differently the time that I did not test it from the time that I did, which leads me to believe that possibly, although I have no positive evidence of the fact, the effect is psychological and not real.

MR. HUCK:—A fleet of trucks in Chicago has shown a great variation in braking. The drivers have claimed that, when the trucks went out in the morning, the brakes were supersensitive, in fact, so much so that they often locked. We made an investigation to find out what was the trouble. We found that occasionally the trucks were washed during the night, that the men washing them turned the hose into the brake-drums, and that some moisture would get in. So long as it was there, the brakes would be very sensitive. The coefficient of friction was at least 20 or 30 per cent higher. There cannot be any comparative data, because the conditions cannot be duplicated. I was wondering whether there is any explanation.

MR. ALLEN:—Although it is purely speculation on my part, a possible explanation might lie in the swelling of the brake-linings due to moisture, thus accomplishing either one or both of two results, one being an actual positive decrease of the clearance, which would be expected to change the braking effect directly, and the other, an increase in the wrapping character of the brake. That is, if the angle of wrap is increased by a given amount and the coefficient of friction remains the same, the amount of decelerating force is not directly proportional to the increase in wrapping but may increase disproportionately as the wrapping angle becomes larger.

MR. STRICKLAND:—I should like to differ with that explanation. The point is the same that I brought up a while ago; after the car has been washed, it starts out wet, begins to dry and gets sticky. You can continue through the stickiness, turn it into a powder and get back to the normal condition. It is merely a question of a thick paste that seizes on the surface of the brake-drum.

CHAIRMAN BREER:—Mr. Strickland has brought out some new points. Mr. Allen's paper primarily shows comparisons under more continued operating-conditions, while most manufacturers and car drivers are interested in what takes place immediately when the brakes are first applied. Many variables enter into this condition.

As Mr. Allen has explained, the expansion of the drums, or of the shoes, especially when there are energizing effects, changes the braking relation. This condition is not necessarily a factor of the coefficient of friction of the brake-lining as much as a change in the mechanics effecting the energizing. In this way, different results are produced which are very noticeable on initial application.

Likewise, the question of mud and foreign matter is one that causes variation. Oftentimes when brake-lining is washed thoroughly, a different degree of braking ability results. This, again, is a factor affecting the lining friction.

Mr. Allen has brought out some interesting comparative results within a range that gives us food for thought, not so much as to the initial application but as to what a given brake-lining really means under continued conditions.

A MEMBER:—My experience has been that the grabbing effect is largely due to the swelling of the lining. That is substantiated by the fact that when a car is left out overnight in a very damp atmosphere, in the rain, for instance, a great deal of condensation probably takes place. On the first application of the brakes in the morning, they grab very badly. When a car is washed, however, is given the same amount of moisture and is taken out immediately, before the saturation has had time to take effect, the same condition cannot be reproduced. It is only after continued standing in the water that the grabbing takes effect.

T. S. SLIGH, JR.[11]:—It occurs to me that, when we consider how rapidly iron rusts after becoming wet, possibly the grabbing after wetting may be attributed to the rusting of the surface of the drum.

CHAIRMAN BREER:—It has been shown that the initial deposit, or slight oxidation of the drum surface, causes a momentary grabbing action.

A MEMBER:—I think Mr. Sligh has pointed out the true cause of initial grabbing. I have been investigating that subject myself during the last year or so in connection with some experimental work. I keep a car on the second floor of a ramp garage and have noticed, particularly after the car has been washed, that in starting down the ramp and in using the brakes, the braking is very severe, but by the time I am on the street, it is absolutely normal.

If you drive along the Atlantic Coast on a muggy day, the humidity does not have to be very high, and you leave the car for a period of an hour or so, you will get exactly the same effect, because the brake-drum has a highly polished surface that is subject to rapid oxidation. I think that is the cause, rather than the swelling of the lining.

A MEMBER:—Regarding the question raised by Mr. Scaife relative to the effect of oil on metal linings, we have a fleet of 16 test-cars on the Pacific Coast on which we have three installations. One of these cars averages about 500 miles a day. Regularly at noon each day the right front brake is greased. Immediately thereafter we run a braking-test on a level concrete floor. At the end of the 24-hr. period, before the car is greased again, we run another test. The drivers of those cars on the two shifts have been instructed to watch for any kind of irregular brake-performance. After 8000 miles of operation, greasing every 500 miles, we have no record of any irregularity of the brake action.

CHAIRMAN BREER:—Is the contact metal-to-metal?

A MEMBER:—Yes.

MR. HOWELL:—A company in this Country has used metal-to-metal contact for 7 years. They have designed the drums so that water is thrown out by centrifugal force; but if oil gets on the metal-to-metal lining, it will not be thrown out, and the only way to get it out is to remove the drums and wash it off with kerosene. You get no braking with oil on the metal-to-metal surface.

J. F. WINCHESTER[12]:—Considerable time has been spent discussing the way in which the brakes were soaked with

[11] M.S.A.E.—Physicist, Bureau of Standards, City of Washington.

water. I should like to have a general expression of opinion as to the types of oil and the methods used in soaking the brake-bands with them.

In the passenger-car industry, on certain types of rear axle, a comparatively light oil is used that frequently gets on the brake-drum. We do not worry much about that, although it does affect their holding ability; but on motor-trucks, where the parts are not so finely built, in many instances a very heavy oil is used to overcome leakage. When this oil gets on the brake-drum, it usually means a lay-up of the truck to take the wheels off to wash out the oil and get back to the original coefficient of friction of the braking material. Has any work been done to determine the relative effects of oils of various viscosities and was it of such a nature that it is of value to the manufacturer and the operator?

MR. HOWELL:—Castor-oil and light engine-oil of about Mobil A consistency will not cause trouble, because they will burn off; but a heavier oil or grease does not burn off and will cause trouble.

MR. ALLEN:—The method of treating the linings with oil was to remove the wheels and thoroughly impregnate or saturate the linings with a heavy engine-oil. In addition, the insides of the drums were thoroughly covered with oil, the wheels were put back, and operation began.

MR. SAFFOLD:—Mr. Copeland of the Johns-Mansville Co. said once that he thought the slipping of the brake, after it had been well saturated with water, was not due so much to the water on the brake-lining itself as to the water that formed a thin film on the brake-drum. He believed that if that film were broken up, the brake would act normally, regardless of the effect on the brake-lining.

CHAIRMAN BREER:—That brings in the question of pressure per square inch of application.

H. D. HUKILL[13]:—Regarding the question of oil on the metal-to-metal linings, I think the problem depends entirely upon the pressure that is brought to bear. Possibly a manually operated brake is not capable of bringing enough pressure to bear to cut through the oil. Under those conditions, a perfectly lubricated brake might, I believe, be compared to a bearing. If enough pressure can be put on it to cut through the oil, the heat will burn the oil off very quickly and the brake will again become effective.

MR. HOWELL:— With heavy oil, a pressure of 120 lb. per sq. in. is not sufficient to cut the oil out.

CHAIRMAN BREER:—The unfortunate situation in the design of brakes is that we are limited, to a certain extent, to the pressure per square inch that can be applied between the lining and the brake-drum. With four-wheel brakes, that is more of a problem than with two-wheel brakes; consequently, with four-wheel brakes, we are introducing many problems that previously did not exist. That is the reason effort is being made to analyze the new fundamentals involved.

MR. HUKILL:—May I supplement my remarks by saying that, in an ordinary metal-brake, the average pressure for emergency application reaches about 200 lb. per sq. in. of lining area?

CHAIRMAN BREER:—Personally, I think both contentions may be right; in other words, two different brake-constructions must be considered. The same thing is true with regard to brake-linings. One lining may work on one design of brakes and not give the same results on another.

[12] M.S.A.E.—Supervisor of motor equipment, Standard Oil Co. of New Jersey, Baltimore.

[13] M.S.A.E.—In charge of automotive division, Westinghouse Air Brake Co., Wilmerding, Pa.

BRITISH CAR DESIGN

IMPROVEMENT generally is the keynote of motor-car design for 1927, greater body-space, greater speed, greater power, more of every desirable attribute in a motor-car. The really small car, the car of diminutive size and also few attributes, with two seats only and without dickey, is conspicuous. The so-called light-car class may, or may not, be light in gross weight, but it must carry a four-seat or saloon body, with upholstery and equipment equalling the luxury cars of pre-war days. The popular size is a nominal 15 hp., developing anything up to 60 b.hp., with a 10-ft. 6-in. wheelbase, 4-ft. 6-in. track, four or five comfortable seats, and capable of 50 to 60 m.p.h.

Superchargers have not so far proved a readier means of increasing power than the usual method of increasing cylinder numbers and dimensions. The cost of increasing the volumetric efficiency in this way is probably as much or more than the cost of increasing the volume of the cylinders.

Four-speed gear-boxes are more numerous than three-speed, but in selecting reduction ratios designers seem to favor a very low first-speed, with the other three steps nearly as widely spaced as in a box providing three speed changes.

Aluminum alloy for chassis frames seems to be the next step in reducing weight, unless the sheet-metal construction, wherein the sides of the body are used as the main part of the beam, is developed in conjunction with the all-metal body. This line of progress seems the logical trend toward making the motor-car an engineering proposition in its entirety.

Wheel patter and wheel wabble still remain, although some attempts are made to damp them out after they have occurred by fitting additional springs. An interesting double steering has been devised, with the elimination of wheel patter as its principal object. The reduction of unsprung weight at the ends of the axles would appear to be the simplest way of eradicating the trouble.

The French design of body wherein the doors are carried below the frame line and the floor sunk in between the main side-members requires cross supports underneath the frame which will need some attention from chassis designers if this type of body becomes popular.—*Automobile Engineer* (London).

MOTOR-VEHICLE REGISTRATION STATISTICS

DATA given in a book entitled The Motor Industry of Great Britain, which was recently published by the Society of Motor Manufacturers and Traders, show that the number of persons per motor-vehicle registered in the United States and Canada is much smaller than the corresponding number in Great Britain, the figures for the three countries being 5.8, 12.0 and 47.8 respectively. However, the number of vehicles per road mile in Great Britain, 5.1, is almost as high as in the United States where the figure is 6.6 and five times as large as in Canada where it is 1.0.

Racing-Car Developments

Duesenberg Describes Two-Cycle Engine and Miller Tells of Front-Wheel Drive in Extension of 1926 Semi-Annual Meeting Discussion

The members will remember that much highly interesting information was given at the Racing-Car Session of the 1926 Semi-Annual Meeting. The discussion was participated in by Fred S. Duesenberg and Harry Miller and many others who have been prominently connected with racing achievements; all under the able chairmanship of Fred Moskovics.

Mr. Duesenberg has been prevailed upon, the members will be glad to know, to enlarge somewhat upon the very informal discussion that he, like the others, presented at the French Lick Springs meeting. What he has to say of two-cycle engine experimentation, as well as other features of racing-car design, is of decided interest.

FRED S. DUESENBERG[1]:—We have had a great many inquiries from different States, from England, and, in fact, from all parts of the world about our two-cycle-engine racing-car. Evidently people have expected a great deal from it. We were rather sorry that we did not have more time to work with it. After we got the first one running, we found that we had had an excessive amount of trouble in starting it. We then tried to develop a way of starting it by putting in an over-running clutch and a starting-motor that would spin the supercharger at some reasonable speed, say, 5000 or 6000 r.p.m., so we could crank the engine by hand. In that case, it would start very readily. But after having been assured by the people making the overrunning clutch that it would operate and give all the power necessary, we found that, when the engine reached the higher speeds, the supercharger evidently took more power and the clutch did not hold. For that reason we were inclined to give up the idea of having any two-cycle engines in the 1926 race.

Comparative cross-sections of the cylinders and valve-actuating mechanisms of the four-cycle and the two-cycle engines are shown in Fig. 1. With the two-cycle engine, it is evident that considerably higher supercharger-pressure is required than with a four-cycle-engine. The speed of the engine is in direct proportion to the pressure of the supercharger. Our gear-ratios on the supercharger had not been laid out for the extreme speeds; consequently, we did not have the high pressure on the supercharger that was necessary to give the maximum speed.

We had to do a great deal of guessing in the work we did on this two-cycle engine. Although I had done some work on two-cycle engines a good many years ago, the conditions were very different; the speeds were not so high. We began with the ports rather narrow. This engine has 2¾-in. stroke and 2 5/16-in. bore. We made the ports ½ in. We began with 7/16 in. The supercharger was running so that we were getting a pressure of about 6½ lb. per sq. in. We could not get more than about 3800 r.p.m., maximum, from the engine, which, of course, we knew was too low. Then we increased the port dimensions to ½ in., and this raised the engine-speed to 4000 r.p.m.

Position of Intake Port

The intake side of this engine is made the opposite of an ordinary two-cycle engine in that the intake port uncovers first and the exhaust port last. The intake port uncovers about ¼ in. earlier than the exhaust port. The rotor is in such a position that, when the intake port opens, the rotor is entirely closed. Then, as the exhaust port operates and is partly uncovered, about 3/16 in., the rotor reaches the point at which it opens the intake port. When the piston rises, the intake port will be open for about ¼ in. after the piston has closed the exhaust port, allowing the pressure in the supercharger to fill the cylinder.

We find, however, that as the width of the port is increased, the efficiency of the engine is increased, and, although we do not have quite so much compression in the cylinder-head, unless we increase the supercharger pressure, we believe that the ports, instead of being ⅝ in. as they are now, should be made about ⅞ in. deep on the exhaust side and about 1⅛ in. on the intake side. I believe it is necessary to get about 15-lb. pressure in the supercharger to get good operation. The engine that ran at Indianapolis had only about 7-lb. supercharger-pressure and was a long way from being anything like an efficient engine. The development work on the engine had not been carried very far and very little work was done to improve its performance after it had been put into the car.

When we had the engine on the block, after we had given it a pretty hard run, we could pick up a spark-plug with bare fingers; it was not half as hot as those in a four-cycle engine. That is a condition we did not expect. Possibly it was because we did not have nearly so high compression in this engine as it would stand.

Trouble with Piston-Rings

The first time we made a run on the Speedway the piston-rings gave some trouble; eight of them were broken. Three pistons in the engine had little pieces pulled out when the rings broke. The pistons began to melt away and we were shooting hot gases into the crankcase, but even that did not seem to stop the engine from running. It seems to me that this type of engine, in its present condition, which is far from perfect, is not as delicate as we had thought it would be. In fact, even though a great many things can be wrong with it,

[1] M.S.A.E.—President Duesenberg Motors Co., Indianapolis.

yet it will run better than a four-cycle engine with only minor things wrong.

I do not know why we did not have trouble with explosions in the crankcase; but we did not have any trouble from that cause. Whether the engine was run 25 to 30 miles after the pistons went wrong or whether it had run only a few miles, I do not know. One thing is sure; the engine is not nearly as delicate as we expected it to be.

As the piston starts moving upward in the cylinder, the rotor continues to turn, and, of course, the opening is very wide when the piston is at its maximum downstroke position and standing still, which gives very good conditions for filling the cylinder; and the duration of port opening is very much less than with the four-cycle engine.

More Sensitive Timing

The timing of the engine is a little more sensitive than we would expect. In opening the intake valve a little too soon, we get popping-back; if we open it a little too late, we do not get the full charge into the cylinder. I believe that can be entirely controlled by the pressure of the supercharger.

Another thing we did with this engine was to put the incoming water under pressure around the spark-plug first, to cool the spark-plug. This did not work well because we found that a steam-pocket formed somewhere in the cylinder, evidently around the exhaust port. The water came in around a sleeve that was around the upper shell that holds the spark-plug. We let the water pass out through small holes in that sleeve, go down

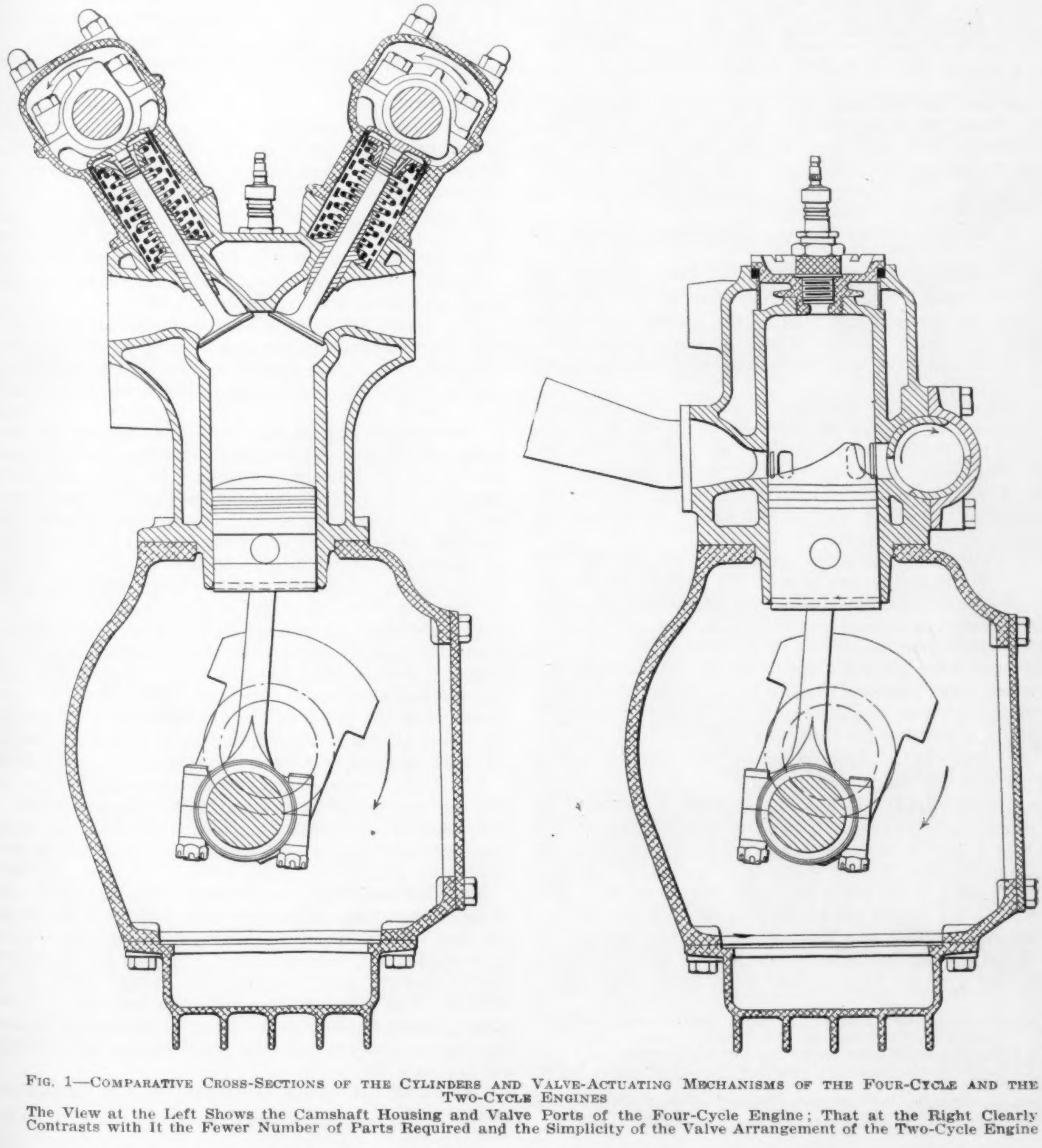


Fig. 1—Comparative Cross-Sections of the Cylinders and Valve-Actuating Mechanisms of the Four-Cycle and the Two-Cycle Engines
The View at the Left Shows the Camshaft Housing and Valve Ports of the Four-Cycle Engine; That at the Right Clearly Contrasts with It the Fewer Number of Parts Required and the Simplicity of the Valve Arrangement of the Two-Cycle Engine

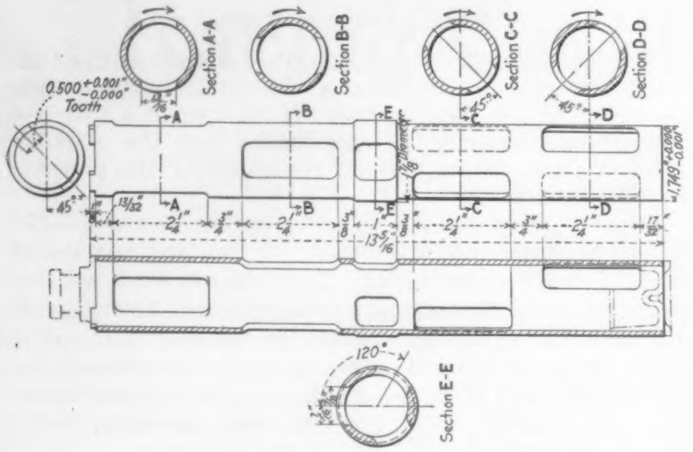


FIG. 2—VIEW OF ROTARY VALVE
The Best Results Were Obtained When the Intake Port Opened After the Exhaust Port Had Opened About 3/16 In. As the Pressure of the Supercharger Increased, a Better Result Was Secured by Opening the Port in the Rotor Later

and around the cylinder and pass around the exhaust port. When we changed that, taking the water around the exhaust port and then around the spark-plug, we found the spark-plug was much cooler than in the four-cycle engine. The engine ran cooler and did not give any indication of having steam-pockets.

V. G. APPLE[2]:—What was the exhaust pressure on the test-block?

MR. DUESENBERG:—I do not know. We did not test that. We made a serious mistake when we laid out this job, because the pressure side of the piston was on the exhaust side and a number of pistons were scored that I think would not have been scored had they been arranged in the opposite way.

We did not discover this until it was too late to change. We intended to change the differential in the rear axle and run the engine backward, but we realized when we were about ready to do so, that we would also turn the rotor in the wrong direction, which would give a bad inlet charge and shoot the gasoline across the piston and through the exhaust port.

Fig. 2 shows the rotor we used when we got the best results. We had only two types. We found that we got the best results when we opened the intake port after the exhaust port had opened about 3/16 in. but as the pressure of the supercharger went up we opened the port in the rotor later and got a better result. We had trouble at first with the engine's tending to spit back; that really was rather serious because there was a large volume of gas in the supercharger that was liable to blow up.

T. J. LITLE, JR.[3]:—Approximately how much power is used in driving the supercharger?

MR. DUESENBERG:—It has always been a guess as to now much power is required to drive a supercharger. A year ago I told a man who wanted to supercharge an 8-hp. engine, that it took about 2½ hp. After he had experimented a little while he wrote back and said that it took 12½ hp. to run the supercharger. For the sake of comparison, I might say that we have a test equipment that is set up for driving a supercharger and are using a 4-in. belt over an 18-in. pulley on a motor that turns at 1180 r.p.m. We had to put on a belt-tightener, to make the belt hold while going over a 4-in. pulley. That 4-in. pulley runs, I believe, around 4200 or 4300 r.p.m. It seems to me that the supercharger takes a good deal of power. I think that perhaps the man was right when he said it took from 12 to 15 hp. Maybe it takes even more.

W. S. HARLEY[4]:—I have run that supercharger with an electric motor. I used 60-amp. fuses with 120 volts.

CHAIRMAN F. E. MOSKOVICS[5]:—What supercharger speed was required to maintain 7-lb. inlet-pressure?

MR. DUESENBERG:—I believe we were running 27,000 r.p.m. at that time.

CHAIRMAN MOSKOVICS:—What horsepower will the four-cycle engine develop?

MR. DUESENBERG:—We have never had one of these engines on an electric dynamometer. We had a fan dynamometer. The only data we get are by comparison. I think Mr. Miller has had some pretty good horsepower readings. He has some hydraulic dynamometers.

CHAIRMAN MOSKOVICS:—How many piston-rings were used in the two-cycle engine?

MR. DUESENBERG:—We first used four rings, in groups of two. Ralph Teetor told me he thought that was wrong. but I thought it would be all right, provided we did not break the rings; but on the first trial eight rings went wrong. Then I called Mr. Teetor up and he said, "We will send over some wider rings," which were the kind he had originally suggested. We have not had any more ring breakage. Evidently that trouble has been cured.

CHAIRMAN MOSKOVICS:—What is the maximum engine-speed obtained, under load and light?

MR. DUESENBERG:—With the supercharger, we had this two-cycle engine running at 4800 r.p.m., with a certain fan load. Two years ago the 122-cu. in. four-cycle engine pulled the same fan at the same setting at the same speed. We apparently got about the same amount of horsepower out of the two-cycle engine, but on the track we did not get nearly the results from it that we got from the four-cycle engine. Probably we did not have the proper gear ratio. When Pete DePaolo took this car on the track, he said: "Your gear-ratio is wrong because, when I come out from the turn, I can run 100 yd. down the stretch and be at full speed." The way the four-cycle-engine car is now, it keeps picking up speed all the way down the stretch.

We did not experiment with many gear-ratios. I found out only yesterday that the gear-ratio of the two-cycle car was 13-46; I thought it was 11-45. The engine ran much slower than it should run. That, of course, means that the speed of the supercharger was much below what it was intended to be; consequently we were not getting anywhere near the power that was intended.

I feel that we should have had a much higher gear-ratio, which would have given us the desired result.

CHAIRMAN MOSKOVICS:—How does the efficiency of the two-cycle racing-engine compare with that of the four-cycle engine?

MR. DUESENBERG:—At present I do not think there is any comparison; the four-cycle engine is much more efficient. But we were surprised at one thing. The two-cycle engine was very economical of gas, which evidently means that we were not shooting enough good mixture into it. The pressure was not high enough. I believe when you get the engine to the point of highest efficiency for power, you are wasting considerable gas that goes out through the ports.

CHAIRMAN MOSKOVICS:—What is the compression ratio?

[2] M.S.A.E.—Proprietor, Vincent G. Apple Laboratories, Dayton, Ohio.

[3] M.S.A.E.—Engineering production consultant, Detroit.

[4] M.S.A.E.—Treasurer and chief engineer, Harley-Davidson Motor Co., Milwaukee.

[5] M.S.A.E.—President and general manager, Stutz Motor Car Co. of America, Inc., Indianapolis.

Mr. Duesenberg:—Every time we filed out the ports we got it a little different. In running these tests, we used a little testing-apparatus that gave about 60 lb. in the four-cycle engine, running at a speed of 4200 r.p.m.; with the two-cycle engine we had only about 46 or 47 lb. The reason was that we could not get the proper amount of space above the piston. In fact, the first time the cars went out on the track the pistons were high enough so that they just touched the head. In four of the cylinders the deflectors were somewhat smashed because they were a little bit too long. Evidently they were so close that when heated they expanded enough to cause contact. When the broken rings got into the combustion-chambers, I do not know how they got in all of them, the tops of the pistons looked like nutmeg graters. I think that that caused some of the trouble from the pistons' heating and melting.

Chairman Moskovics:—How far do the exhaust ports extend around the cylinder?

Mr. Duesenberg:—About one-third of the way.

Chairman Moskovics:—Has the Roots blower been tried?

Mr. Duesenberg:—No.

Chairman Moskovics:—It might be of interest to know that no American racing-car is using the Roots blower-type supercharger. Practically all use the General Electric supercharger.

Mr. Duesenberg:—I believe that the Roots-type blower would be much more satisfactory for the two-cycle engine because it is very hard to turn this engine over fast enough to get pressure from the turbo-blower. This is particularly true because we use a rotor that holds the port open as the piston goes up, and a little of the mixture will blow back through the carbureter, which makes it hard to get a good charge into the cylinder.

Chairman Moskovics:—How is the engine-speed controlled?

Mr. Duesenberg:—By the carbureter throttle.

Chairman Moskovics:—What is the minimum speed, idling?

Mr. Duesenberg:—We have not had very good results at the lower speeds because we get practically no supercharger pressure below 2000 r.p.m. The engine will run as slow as 1000 r.p.m. but the mixture is hard to control; it is almost impossible to get the engine to idle nicely and to get a clean-burning mixture.

W. G. Wall[6]:—Do you notice any material difference in the compression?

Mr. Duesenberg:—The compression was considerably lower in the two-cycle than in the four-cycle engine; naturally, it would stand more spark-advance. It was necessary to carry the spark pretty well advanced in order not to have backfiring; if the spark were retarded much it would backfire very easily.

L. W. Oldfield[7]:—Mr. Duesenberg has given the port heights and has stated that the exhaust port extended approximately one-third of the way around the cylinder. Is that also true of the inlet port?

Mr. Duesenberg:—It is.

Mr. Oldfield:—I do not remember whether you gave the calculated compression-ratio.

Mr. Duesenberg:—We originally laid it out to get about 4½ to 1. The four-cycle has about 5½ to 1.

Mr. Oldfield:—I heard at the track that you were firing two cylinders as one and using a standard crankshaft.

Mr. Duesenberg:—Nos. 1 and 8 fire at the same time, and correspondingly throughout the block. We did not think that it was the proper thing to do, but we felt that the two-cycle engine was so far "up in the air" that we should probably have to run the engines as four-cycle engines. The whole job was laid out so that within 3 or 4 hr. it would be possible to change over the two-cycle into a four-cycle engine and use the same crankshaft. We did not have time to make two sets of cranks and to be sure that we might have some engines that would run, we decided to fire the two cylinders at the same time and use the same crankshaft.

Mr. Oldfield:—You say you fired No. 1 and No. 8 together?

Mr. Duesenberg:—Yes.

Mr. Oldfield:—Then your present crankshaft design is different from your former crankshaft design?

Mr. Duesenberg:—Yes.

W. L. Carver[8]:—It is Nos. 1 and 8, 3 and 6, 4 and 5, and 2 and 7 simultaneously respectively.

Mr. Duesenberg:—I cannot remember the figures. We have two or three different shapes of crankshaft but we tried only one type.

L. V. Smith[9]:—What advance do you use on the two-cycle engine? In other words, have you found any difference between strictly magneto ignition and the battery type?

Mr. Duesenberg:—We have not tried magneto ignition on any of these engines to date. We have two engines on which magnetos are mounted, but they have not been out on the Speedway. The advance that we are using on the two-cycle engine is about 45 deg., while on the four-cycle engine we usually use from 35 to 38 deg. That depends largely on the compression. The compression of our present four-cycle engine is not nearly as high as it should be. When we were ready to put the engines together we found that the piston bosses were too high and that we could not get nearly the compression that we originally intended to have.

W. W. Bertram[10]:—What types of spark-plug were used?

O. C. Rohde[11]:—We have reduced the design of spark-plugs for racing-cars almost to an exact science. In past years it was guess-work. Now we can almost, not quite, take the specifications of a particular engine and design a plug to meet the conditions of that engine. Within a short time we hope to be able to do it exactly. We have hit it exactly a couple of times. The plug used in the Duesenberg two-cycle engine was what we call an R-11 special. It was a plug specially developed for the race this year and designed to run exceptionally cold. When Mr. Duesenberg spoke of the plugs in the two-cycle engine being cool, it was partly because of the wonderful cooling-system he has worked out and partly because we gave him a plug that would run cool rather than hot, as we usually run them, because we knew that in the two-cycle engine we should not be troubled with oil and could afford to run the plugs cool without danger of fouling them.

Plugs in racing-cars are not just spark-plugs. When the operators take the plugs out of the engine and look at them they are checking the mixture ratio, because the plugs have become an indicator of the mixture in racing-engines. There are not many persons who can tell, by

[6] M.S.A.E.—Consulting engineer, Indianapolis, Ind.

[7] A.S.A.E.—Consulting engineer, in charge of design and development, Pac-Age-Kar Corporation, Chicago.

[8] M.S.A.E.—Field editor, Chilton-Class Journal Co., Philadelphia.

[9] A.S.A.E.—Mechanical engineer, Charles McCaul Co., Philadelphia.

[10] M.S.A.E.—New York district office manager, Morse Chain Co., New York City.

[11] M.S.A.E.—Chief engineer, Champion Spark Plug Co., Toledo.

looking at the plug exactly what the mixture is, but those of us who know can tell by looking at the color of the insulator at the end of the plug just what the mixture is.

Color of Spark-Plugs an Index of Mixture

A very white insulator means too lean a mixture and danger of burning the valves; too dark a color on the insulator means a rich mixture that will have a tendency to foul the plugs in a short time. Just right is a color half-way between, which is rather hard to designate; it is a sort of tan or buff color. It is really surprising how accurately those who understand can gage mixtures by examining the spark-plugs.

I have a sample kit of our racing types of plug. You can see what the idea is in connection with racing-plug design. These plugs range from the coldest plug to the hotter types for the more ordinary engines, with a few specials mixed in between. They begin with the R-2, which was designed especially for Mercedes racing-engines, and also runs successfully on the P. F. two-cycle engine, which, according to the reports, is running 11,000 r.p.m.

The next is the R-11 which was designed specially for board-track racing-work in the Miller and the Duesenberg cars where the heat becomes excessive and the compression pressures in some cases are a little higher than they should be. At the Indianapolis race, there were a few cars that had excessive compression-ratios and ran very hot. That plug had to be used in them. This particular plug, with a little trick arrangement, was the one that was used in the two-cycle engine and showed very clean and cold after a hard run.

The R-1 has been the Standard Speedway racing-plug for the last 3 years. It was used on practically all the Miller cars in the race this year. On board tracks we may have to go to the colder plugs.

Then come the R-7, R-6 and R-5, which we have developed primarily for aviation work; the R-8 which was developed the same as the R-2, except for the longer shell, which brings it out of a pocket in the engine; also the R-9 which is used in the motorcycle and the Liberty-engine fields. Then there are some specials which were designed for European racing work where there was a tendency to "oil-up" more and the plug came entirely through the cylinder-head instead of being slightly pocketed, the way the plugs are on the Miller and the Duesenberg engines. In these plugs, we simply pocketed the spark in the plug itself so as to get approximately the same condition.

PROF. H. A. HUEBOTTER[12]:—How do the superchargers and the carbureters compare between the 2.0 liter (122-cu. in.) and the 1.5-liter (91-cu. in.) engine and between the two-cycle and the four-cycle engine?

MR. DUESENBERG:—The first test was really the only test I made on the two-cycle engine. We used exactly the same size supercharger that we did on the four-cycle. We expected that we should have to use a larger supercharger on account of taking a charge every revolution, but we found that, with the port openings and the conditions we had, it would mean a drop of pressure of only ¼ lb. when using both blocks, as compared with using one block, which proved to me that we had a supercharger that was plenty large enough. In fact, it is larger than is necessary the way things are at present, with the present amount of port-opening. When we make the port-opening probably one-third greater than it is now, the supercharger ought probably to be slightly larger. I am not sure of that.

PROF. HUEBOTTER:—Did you use the same supercharger on the four-cycle engine?

MR. DUESENBERG:—I believe we did.

MR. OLDFIELD:—We noticed that the two-cycle engine smoked slightly all the time. Was any lubricant used in the fuel or was the oil used exclusively to lubricate the rotor?

MR. DUESENBERG:—The smoke seemed to be caused by the lubricant on the rotor. The grooves are a little too wide. We have developed another type of rotor, in which the grooves are very narrow, that does not smoke nearly so much. We do not use oil-rings on the two-cycle engine. We used simply straight compression-rings and, so far as we can tell at present, the smoke was caused entirely by the lubricant on the rotor, which had too much pressure on it.

MR. OLDFIELD:—Is any lubricant used in the fuel?

MR. DUESENBERG:—No . We thought it might be necessary, but after test we found we had ample lubrication of all parts.

CHAIRMAN MOSKOVICS:—How does the fouling of spark-plugs on the two-cycle engine compare with that on the four-cycle?

MR. DUESENBERG:—The spark-plug situation has always been a rather delicate one. Of late years, however, the spark-plug makers have kept a little closer check on the situation and, while they might come down with 2 or 3 bu. of plugs before the race, they would hold back a few of them. I can remember, not so many years ago, when some of the star drivers adjusted the carbureters. Whenever they made a carbureter adjustment they had to put in a new set of plugs to meet that adjustment and in doing so they used up 50 or 100 plugs before they succeeded in adjusting the carbureter. About that time a valve would break or something else would go wrong and they would have to start over again. By the time the race started, there would not be many real good spark-plugs available.

R. R. TEETOR[13]:—What was the oil-consumption of the two-cycle engine compared to that of the four-cycle engine?

MR. DUESENBERG:—We did not keep a record of the oil-consumption. I have not even asked how much was used, but naturally the oil-consumption on that job must have been rather high.

MR. CARVER:—Has Mr. Duesenberg experimented with putting the rotor in the head instead of along the side?

MR. DUESENBERG:—We have not done so yet, but a few sketches have been prepared in which we have planned to use a rotor in the head and exhaust ports on both sides of the cylinders. I believe that is a good idea. I believe the rotor in the head can be made successful, if a proper oil-seal can be obtained when running at the higher speeds, as I am sure an engine of that type would run; the seal should be very good. I believe that having the rotor in the head will tend to make it hot enough to cause trouble. It would make a much better engine and the piston would be much lighter.

CHAIRMAN MOSKOVICS:—It would do away with the deflector.

MR. DUESENBERG:—With the intake port as high as it should be to get proper operation, the deflector must necessarily be extra high. It means that we must have a large area of surface that is bound to get hot and considerable weight to give the proper strength, so that the piston is bound to be heavy.

---

[12] M.S.A.E.—Research associate, professor, Purdue University engineering experiment station, West Lafayette, Ind.

[13] M.S.A.E.—Chief engineer, Indiana Piston Ring Co., Hagerstown, Ind.

Another thing, the deflector would overheat if it were not for the fact that the gas under considerable pressure is shot against the deflector and cools it rapidly so that, while one may offset the other, the arrangement still has many advantages. From the fact that the two-cycle engine seems to run much cooler than the four-cycle, I imagine that not quite so much water would be used.

A. W. HERRINGTON[14]:—No water was put into that car at any time during the practice or during the race.

MR. CARVER:—What do you think of the flexibility of the supercharger type of two-cycle engine, that is, its adaptability to ordinary commercial uses?

MR. DUESENBERG:—With the present type of supercharger, such as we are using, the flexibility of the engine is nil; it has none; but I believe that, with the three or four different types of supercharger that are being developed along the lines of the Roots-type blower, that feature can probably be regulated so that it will be all right.

PASSENGER-CAR ENGINE-SPEEDS

MR. LITLE:—Do you believe that the passenger-car engine of the future will be a high-speed engine of this type?

MR. DUESENBERG:—That is a hard question. I hardly think so; yet it is quite possible on account of the absence of valves and the many other things that usually cause trouble on cars. I was very much surprised at the way this engine ran after we took out the narrow rings. The cylinders were scored a little and many little things happened, but after we had repaired the engine and had put in new pistons and wider rings, it ran very regularly and has been gradually improving although the pistons were a misfit, because some of the boys, not following instructions, put some pistons into the engine that were 0.014 in. too small. They did not have time to make another set, so there were three pistons in the engine that had a clearance of 0.014 in. and five others that had a clearance of 0.008 in.; but it seemed to run just the same.

CHAIRMAN MOSKOVICS:—We are all trying to get lessons from this for our everyday life. About a month ago I received a letter from one of the foremost engineers of Europe, who up to the present has been addicted to or afflicted with high-speed engines. He is bringing out a new car. The car has over 800-cu. in. displacement, 2000-r.p.m. maximum speed and two gears; the top speed will be 124 m.p.h., the lowest, 85 m.p.h. for passenger cars. The subject seems to be about as open as a thing can be.

MR. DUESENBERG:—I do not believe that the present-day high-speed racing-engine is the type that will be followed by engineers for passenger-car service. I do believe, however, that if the valve mechanisms and the valve-operating mechanisms can be brought down to a point at which they approach somewhere near to the present-day racing-engine, the speeds of the passenger car can be increased about 25 or 30 per cent and will give better efficiency; but it is the valves and the valve mechanisms, as I see it, that will have to show most of the improvement.

CHAIRMAN MOSKOVICS:—Bear in mind this is all relative; when Mr. Duesenberg talks of racing-car engines, he is talking of 7000 r.p.m. He would call a 4000-r.p.m. engine a moderate-speed engine.

MR. DUESENBERG:—Our engines are not running 7000 r.p.m. and I think the speed of a passenger-car engine at the normal speed that a car is allowed to run, say 40 m.p.h., will be far below 2000 r.p.m. for some time to come.

C. A. TRASK[15]:—What metal do you use for the valves?

MR. DUESENBERG:—We have used some silchrome steel. We leave that to the steel people. We think they ought to know more about it than we do.

EIGHT-CYLINDER CRANKSHAFTS

MR. OLDFIELD:—I should like to ask Mr. Miller the same question I asked Mr. Duesenberg. So far as I know, all Mr. Miller's cars have the crankshaft made as two fours, twisted in the middle. Is there a definite reason for that and has he ever made tests of the same engine otherwise than with the two different designs of crankshaft?

H. A. MILLER[16]:—I have never experimented with any other type of crankshaft.

MR. DUESENBERG:—We find that in our engines when the crankshaft is made with two fours, we do not have so smooth-running an engine at the lower speeds as we have with the other type of shaft. I do not know what the condition may be at the higher speeds, because we have not gone up to the top yet and, besides, we have had only one engine that has been driven at the higher speeds.

MR. OLDFIELD:—I think it is important that we find out why Mr. Miller and Mr. Duesenberg have been so successful with this crankshaft, while all the builders of eight-cylinder engines refuse to consider it as correct. We must admit that Mr. Duesenberg and Mr. Miller have achieved results; there is no question about that. If they get such wonderful results, we should like to know why.

MR. HERRINGTON:—The carburetion problem may be serious when there is no supercharging; but we all have been more or less "suspicious" of the fact that the supercharger has eliminated a great deal of distribution trouble. Possibly that introduces a variable that must be taken into consideration in making any test on the two types of shaft.

CHAIRMAN MOSKOVICS:—Mr. Miller says that, originally, the problem was one of carburetion, but that now, with the supercharger, they have not had an opportunity to make a test. The company that I am familiar with went through the same mental difficulties and the 2-4-2-shaft decision was based on mathematical balance more than anything else. The supercharger might change the reason for using the ordinary 2-4 shaft.

Mr. Miller's engines run about 7000 r.p.m. as follows: 7000 r.p.m., 154 hp.; 6500 r.p.m., 150 hp.; 6000 r.p.m., 136 hp.; and 5500 r.p.m., 118 hp. The bore and stroke were 2-3/16 x 3 in.; eight-cylinder; compression ratio, 5.35. It has a supercharger, of course.

It happened that one of Earl Cooper's cars tried to climb the fence unsuccessfully a few days before the race. We had to make some of the parts in our experimental room. Mr. Miller came here without a spare part. Cooper said there was no comparison between the cars. He had also driven one of Mr. Miller's cars previously and could make the turns just as fast. He was very enthusiastic. I might also say it is the prettiest piece of workmanship I ever put my eyes on. That front-wheel design was wonderful.

SUPERCHARGERS

Would you like to know the speed of the supercharger at the various speeds? The peripheral speed at the tip

[14] M.S.A.E.—Consulting automotive engineer, City of Washington.

[15] M.S.A.E.—Equipment efficiency engineer, Rockwood Mfg. Co. Indianapolis.

[16] M.S.A.E.—Owner, Miller Engine Works, Los Angeles.

of the blade of the impeller is 68,533 ft. per sec. at 7000 r.p.m. That is equivalent to 720 m.p.h.

MR. APPLE:—What supercharger pressure does that represent?

MR. MILLER:—It represents 16½ lb.

CHAIRMAN MOSKOVICS:—This is a supercharger that Mr. Miller developed himself. One peculiarity that I notice is that the blade is what is called a bucket blade, made of vanadium steel.

MR. MILLER:—All previous impeller blades have been made of an aluminum alloy at our foundry. Lately they have been made with a piece of drawn vanadium steel, not forged. We had two of them made at the Stutz plant and both broke because they had not been heat-treated. We had trouble with some of them breaking off at the collar, and thought it was due to a flaw in the casting; but, on looking over some of them after the race, we found that little pieces of rock had flown up and hit them, bending them a little. As they were turning over at a very high rate of speed, one of them was bent and thrown out of balance. They break off and go through the supercharger and not much is left of them.

CHAIRMAN MOSKOVICS:—The rate of impact was 680 m.p.h. It does not take much of a particle to do damage at that speed.

MR. DUESENBERG:—Has Mr. Miller had any trouble with the blades or wheels getting larger? One man did not get into the race. He came here from San Francisco and had some parts for his car made up in Indianapolis some time ago. He came into our plant and said: "I am having trouble with my supercharger impellers breaking and stretching."

I said: "What speed are you turning?"

"About 40,000 r.p.m. I have tried three different kinds and whenever I get my car up to speed something happens. When we measure the impeller wheel, after running for 10 or 15 miles, we find that it has grown about ⅛ in."

CHAIRMAN MOSKOVICS:—I want to tell you that that is some of the prettiest machine work I have ever seen. The blade is slightly undercut. Mr. Miller called attention to the fact that the little buckets we made for him were made of rolled stock; they were not cast. He felt that either the cast or chilled forged stock would have been better. They are just like a watch to work on.

M. H. THOMS[17]:—What is the diameter of the rotor?

CHAIRMAN MOSKOVICS:—The diameter of the rotor is 7 in.

MR. THOMS:—I mean the bucket itself.

CHAIRMAN MOSKOVICS:—About 3 or 4 in.

QUESTION:—What is the car speed corresponding to the engine-speed? At the Los Angeles Speedway at 6400 r.p.m. he got 130.8 m.p.h. with the gear-ratio used at Beverly on a board-track. On varying tracks varying ratios are used. The same gear-ratio could not be used at Indianapolis because the track is four-cornered with a much harder turn.

FRONT-WHEEL DRIVE

G. T. BRIGGS[18]:—What is the future of the front-wheel drive?

MR. MILLER:—The front-wheel drive has great possibilities. It pulls the car instead of pushing it, which is a very good thing on dirt roads. It seats passengers very much lower and is very much easier to drive. It does not steer hard; we have no trouble at all with the steering. All the mechanism is out in front so that it can be reached very easily. It is a little more expensive to make at present, but I do not think it would be in production.

CHAIRMAN MOSKOVICS:—It is no particular secret that the Packard Company bought one of Mr. Miller's cars last year. One of Mr. Vincent's criticisms was that it took a little longer wheelbase than the normal construction. On the other hand, the rear spring is not complicated by any stresses. It is free to act as a spring. The car can be made any height, which, of course, gives all the well-known advantages of low construction. Mr. Miller says that the reason, possibly, that the Packard layout was 10 in. longer was that they tried to incorporate it with their existing type. He says the wheelbase need not be longer.

MR. LITLE:—Inasmuch as we hear that Fiat is intending to run at 11,000 r.p.m., I wonder whether we can expect some increase in engine-speed next year.

MR. MILLER:—I do not feel we have yet reached the limit in speed. I agree with Mr. Duesenberg that, up to the present, the valve action would enable a greater engine-speed to be obtained, if the valves did not have to start and stop so many times.

JOHN YOUNGER[19]:—Has any experimentation been done on worm-drive axles for racing-cars?

MR. DUESENBERG:—We have not done any actual work on worm-drives. About a year and a half ago we worked with a drive that was not exactly a worm-drive or a bevel-drive, but a combination of both.

CHAIRMAN MOSKOVICS:—Was it a hypoid gear?

MR. DUESENBERG:—No. The job was not very satisfactory although the gears did show wonderful possibilities. The possibilities of cutting them have not been such that we could get a perfect worm. We have had one of those axles made up for more than a year, but it has never been put under a car.

CHAIRMAN MOSKOVICS:—Earl Cooper brought about five sets of gears. You can instantly see the impossibility of getting that worm 98-per cent efficient; that would mean some pretty fine work and would cost many thousands of dollars because the gear-ratios must be shifted. They may be shifted because of a heavy wind. On the front-wheel drive, there would be no reason for a worm, because the car is down as low as they can get it anyway. The real advantage of the worm drive is in lowering the frame and still getting ample clearance. Such things do not cause worry now.

E. D. HERRICK[20]:—What valve diameter and clearance does Mr. Miller use?

CHAIRMAN MOSKOVICS:—In the clear, 1 3/32 in. The spring pressure is 110 lb. open and 87 on the seat; the lift is 5/16 in.

MR. HERRICK:—What horsepower is required to drive the supercharger?

MR. MILLER:—At the usual speed, about 18 or 20 hp.

CHAIRMAN MOSKOVICS:—There would be some ratio downward. Last year we had about 14 hp. at about 5500 r.p.m. On the 122-in. cars, the maximum engine-speed is about 6000 r.p.m. In this race several cars were the regular 122-in., changes being made only in the crankshaft. Mr. Duesenberg's cars did that, as well as several of Mr. Miller's cars.

C. M. MANLY[21]:—What was the size of the crankshaft journals?

[17] M.S.A.E.—Supervisor of inspection, Marmon Motor Car Co., Indianapolis.

[18] A.S.A E.—General sales manager, Wheeler-Schebler Carburetor Co., Indianapolis.

[19] M.S.A.E.—Editor and publisher, *Automotive Abstracts;* professor, industrial engineering department, Ohio State University, Columbus, Ohio.

[20] M.S.A.E.—Assistant chief engineer, Lycoming Motors Corporation, Williamsport, Pa.

[21] M.S.A.E.—Consulting engineer, Manly & Veal, New York City.

MR. MILLER:—They were $1\frac{5}{8}$ in.

CHAIRMAN MOSKOVICS:—What were yours, Fred?

MR. DUESENBERG:—They were $1\frac{5}{8}$ in.

MR. CARVER:—How would the universal-joints on the front-wheel drive act in road work? In racing, the angularity of the front wheels in making turn is relatively slight.

MR. MILLER:—The universal-joints on a front-wheel drive travel at wheel speed, whereas on a rear-wheel drive they travel at engine-speed. The front-wheel drive has very little movement up and down, only about $1\frac{1}{2}$ in. from one extreme to the other. The universal-joints are equipped with ball-bearings and last indefinitely.

MR. CARVER:—Do you overcome the variation in angular velocity between the driving side and the driven side of the joint?

MR. MILLER:—No. Going as slow as they do, we do not notice it.

CHAIRMAN MOSKOVICS:—The conditions at the Indianapolis Speedway are more nearly like those of a road than are those of any other course. There are a good many bumps in it; a car that will stand up under those conditions will be pretty good on the road. Those who saw the races noticed the drivers on the home-stretch leaving the ground; the bumps on the back-stretch were still worse. They probably had all the action of ordinary road conditions.

MR. MILLER:—Another point that it might be well to note is that in steering the front-wheel-drive car, we do not have to over-steer as we do with a rear-wheel drive.

CHAIRMAN MOSKOVICS:—I went over to the track this year and noticed the various cars coming round. Both front-wheel-drive cars went around the corner without a move; they were simply in a groove. That is what they call it. You could see every one of the other cars twist as they over-steered and straightened out.

MR. LITLE:—It is easier on the tires, is it not?

CHAIRMAN MOSKOVICS:—Much. After the race was about 1 hr. old, the track to a width of 10 ft. was flooded with oil on the turns. Every skilful driver tried to keep his car in that oil. The track was like a grindstone. The whole trick was how far a driver could keep the car down in the oil. Neither of the front-wheel-drive cars ever left the oil so long as they were in the race. They never had to straighten up. All the other cars had one wheel from 6 in. to 2 ft. out and were grinding the tires.

F. M. YOUNG[22]:—How does Mr. Miller get his drive for the supercharger? From what does he drive it? Also, what metal does he use in the journals?

MR. MILLER:—The supercharger is driven from the crankshaft between the flywheel and the rear main bearing.

CHAIRMAN MOSKOVICS:—You have an intermediate gear to get up to speed, have you not?

MR. MILLER:—Yes.

CHAIRMAN MOSKOVICS:—This gear is driven from the crankshaft. The old ones were driven from the camshaft. That was really a makeshift because they put superchargers on the 122-in. engine. They have not had any trouble with the gears on this engine.

MR. MILLER:—We had some trouble with the gears, but it was owing to the design. After the race, when we took the cars down, we found we had a double gear driving two sides and half the gear split. It slipped on the shaft, when the engine got hot, threw all the work on one gear and caused some of the teeth to break off. The material in the connecting-rod big-end is just plain babbitt made by the Syracuse Refining Co.

MR. MANLY:—What was the clearance on the $1\frac{5}{8}$-in. main journals?

MR. MILLER:—It was 0.0025 in.

MR. DUESENBERG:—What was the outcome of the overrunning clutch that Mr. Miller tried out? Did he use it in these cars?

MR. MILLER:—I made several experiments with the overrunning clutch and think it could be used all right if it were properly designed. The incline was not at the right angle, and at certain speeds it took certain power; the clutch would slip.

MR. DUESENBERG:—Do you feel that it is necessary or advisable to have an overrunning clutch?

MR. MILLER:—I do not think it is necessary.

MR. DUESENBERG:—I felt that the overrunning clutch was not necessary on ordinary superchargers. The engine does not slow down as quickly as it picks up. I have been afraid of the overrunning clutch but used it on the two-cycle engine so that we could put a starter on and get up some supercharger speed without having the engine turn over. The clutch we used, although it was calculated to have seven or eight times the amount of pulling capacity that we needed, did not hold. Instead of getting the 7 or 8-lb. supercharger-pressure that we expected when it was hooked-up solid, we had only 3 or 4 lb.

MR. MILLER:—I had about the same experience, except that the pressure was a little higher. The clutch will slip at a certain pressure.

MR. DUESENBERG:—Do you think that is caused by the high engine-speed, on account of the vibrations set up by the engine?

MR. MILLER:—I do not think it is caused by the vibrations.

MR. DUESENBERG:—That is what we thought.

MR. MILLER:—I think that, if we had a wider surface and the right incline, it would work all right. The little engines do not slow down as quickly as the large ones, and that gives the supercharger a chance to slow down. At Indianapolis this year when the driver came to a corner and took his foot off the throttle, the car would go just about the same, so he had to keep taking his foot off a little farther back, whereas with last year's cars, the engine would slow down very quickly.

CHAIRMAN MOSKOVICS:—The result is there is less strain put on the supercharger than last year.

MR. DUESENBERG:—Is that due to the fact that we have reduced the piston diameters and the stroke, yet have not reduced the moving parts very much on the 122-in. engine?

CHAIRMAN MOSKOVICS:—The cars are the same weight as they were last year, with the same frontal areas.

MR. MILLER:—I think it is on account of the small-size piston and smaller parts.

JOHN MCGEORGE[23]:—I would like to ask about the steering of the front-wheel drive. So far, the Ackerman steering-gear has been used, and that is limited to about 30 or 35 deg. of steering-angle. It is possible that a front-wheel drive uses a much larger angle of steering. Has any effort been made to develop a steering-gear to utilize that?

MR. MILLER:—We make our own steering-gears. As we have never used the front-wheel drive commercially, we have never had occasion to use the steering as we would on regular rear-wheel-drive cars. The steering-angle is about 30 deg. on the racing-car that we had at

[22] M.S.A.E.—Vice-president and general manager, Racine Radiator Co., Racine, Wis.

[23] M.S.A.E.—Technical adviser on special engineering work, Oakland Motor Car Co., Pontiac, Mich.

Indianapolis; and we never use more than 3 or 4 deg. in going around the track. The ordinary rear-wheel-drive requires considerably more steering because the front wheels slide instead of going where they are steered.

W. R. STRICKLAND[24]:—What is Mr. Miller's experience with the two-cycle engine?

MR. MILLER:—I spent several years experimenting with a two-cycle engine beginning about 20 years ago. I had difficulty in making it run slow. We reached 3000 r.p.m. that time in an engine of about 20 hp. I think the possibilities of the two-cycle engine with the supercharger are very good.

MR. STRICKLAND:—I understood that you actually built some straight eights with two-cycle engines and then changed them.

CHAIRMAN MOSKOVICS:—One of Mr. Miller's cars came to the track with a two-cycle engine but he had nothing to do with it. One of the drivers had had some outside company build it. It did not start in the race, as I remember.

MR. MILLER:—That really was a two-cycle engine with valves; it was the valves that caused trouble. I have never seen it but I know there was considerable trouble with the valves. They had to work twice as fast as in an ordinary engine and it was hard to keep them all in one piece.

CHAIRMAN MOSKOVICS:—Captain Rickenbacker told me he was very much encouraged, in fact, very enthusiastic on the subject of two-cycle engines with superchargers.

MR. YOUNG:—Did Mr. Miller change the capacity of the radiator for the cooling-system when he changed the engine from 120 to 91½ cu. in?

MR. MILLER:—It was changed. We have better cooling now than we had before. At present we do not use any louvers in the hood. The reason for changing is that we do not use any shell on the outside of the radiator tank. The air comes into contact with the radiator shell, which is the tank, and cools it much more than it would if a regular shell were over it. It is a little more expensive to make but is much more durable. The core was of the same depth and about 1 in. shorter.

CHAIRMAN MOSKOVICS:—He had the cutest little auxiliary radiator cast in the aluminum shell on the bottom that I ever saw. I think he had it there because it was convenient.

MR. MILLER:—I put it there to get the water out.

MR. YOUNG:—How was the aluminum tank soldered?

CHAIRMAN MOSKOVICS:—It was not soldered; it was bolted down.

MR. MILLER:—It was just for convenience in getting a water connection for the pump and making a better-looking bottom.

J. E. HALE[25]:—It seems to me that there must be rather more wave in the front wheels of the front-wheel-drive car as compared with other cars. Will Mr. Miller tell us what designs he incorporated in this car so that it would not shimmy?

MR. MILLER:—We put all the weight we could on the front wheels of the front-wheel-drive car. I do not know exactly how much that is. On the conventional car the weight is very nearly equal on all four wheels. There is about 120 lb. difference in the rear-wheel drive, but on the front-wheel drive I neglected to weigh it. All we have on the back is the tank. The rear axle is very light; it weighs only about 26 lb.

MR. HALE:—I should have mentioned that I was interested in the unsprung weight, because it seems to me that unsprung weight is one of the factors that has to do with shimmy.

MR. MILLER:—We have very little unsprung weight in the front-wheel drive. Owing to the spring-suspension, we have only about 20 lb. more unsprung weight on the front axle than we would have on a conventional-type car. The transmission is sprung, is fastened directly to the frame; all we have is a bent axle in front of the transmission. The brakes are also sprung; they are on the transmission instead of in the wheels. I have made them both ways and prefer to have them sprung rather than unsprung. There is no tendency to shimmy if the wheels are balanced.

MR. DUESENBERG:—I forgot to mention when I was on the floor that about 8 years ago we built an engine for the Government for a tractor job. It was a four-cylinder engine that had 1200 cu. in. When it was tested, Mr. Miller sent Dave Lewis over to fit the carbureter. At that time at 1500 r.p.m. we got 205 hp. The man who was looking after the job thought that was very remarkable. We had a 2-in. Miller carbureter on it that seemed to act very well.

While I was in Los Angeles on Thanksgiving Day, 1925, Mr. Miller gave me a test sheet of a run of a 120-cu. in. engine just 1/10 the size of the one we had tested several years before. I believe he also used a 2-in. carbureter. That engine developed 203 hp. In 8 years there has been a gain of 900 per cent. That is pretty good development.

MR. TEETOR:—With regard to piston-rings in a racing-engine, no unusual radial compression is applied. About the only thing that is necessary is to apply a slight wall-pressure uniformly and make the ring very narrow so as to cut down the friction as much as possible. The oil problem, as we understood it until a few years ago, was considerable because of the fact that, when the engines were throttled down or shut-off for the turns, the oil was thrown into the cylinders. It is well known that an engine that is developing less pressure in the combustion-chamber will pump oil much more freely. When the engine is shut-down on the turns, an excessive amount of oil passes the pistons, and when the engine is opened up again the oil burns and smokes. This has been greatly reduced by the use of the slotted-groove type of oil-ring. We think that combustion has been improved by not throwing a heavier oil into the combustion-chamber. Just what effect this has I cannot say. If tests have been made on engines equipped with oil-regulating rings and not equipped with oil-regulating rings, they will give you more information on that subject than we can.

As Mr. Duesenberg has said, he did not feel that the two-cycle engine needed oil-rings; it did not seem to us that it did. I have had very little personal experience with racing-engines. Most of the information we have been able to gather has been from contact with Mr. Miller's and Mr. Duesenberg's organizations.

MR. HARLEY:—We have raced for many years with motorcycles. Although the motorcycle is different from the automobile, the problems are somewhat the same. We are turning engines out now that run 7000 r.p.m. We are building the little engines at this time with overhead valves, the general type of the racing-engine. The engine has 21-cu. in. displacement and gives 21 hp. at 7000 r.p.m. We use roller-bearings throughout the engine and have done so for years. We tried to use a supercharger. Mr. Duesenberg told me it took 2½ hp. I spent $2,000 building one and when I put it on the motorcycle

[24] M.S.A.E.—Assistant chief engineer, Cadillac Motor Car Co., Detroit.

[25] M.S.A.E.—Manager of development department, Firestone Tire & Rubber Co., Akron, Ohio.

I found out it took 12 hp. We do not get the results with superchargers that automobile-engine designers do because we have an additional problem on account of surging because of the single cylinder. I do not think that the type of supercharger that is used in the automobile is suitable for the motorcycle. Motorcycle high-speed work is somewhat passé in this Country, but we need it abroad. We have to race against the British all over the world.

CHAIRMAN MOSKOVICS:—The use of "doped" fuels has made a marked change in the whole situation. I will give you one little illustration. Three years ago the Bugatti team came over and I suggested using ethyl gasoline. Tom Midgley said, "Find out what the compression-ratio is." They said it was 7.3 to 1. Tom sent one of his assistants over with some ethyl gasoline and the engine lost 250 r.p.m. That surprised us. We found that the compression-ratio was 6.25. When the dope was made for that compression-ratio the engine gained 150 r.p.m. over its original speed. I am not being subsidized for saying that the winner of the race this year used ethyl gasoline.

THOMAS MIDGLEY, JR.[38]:—I will give you a brief history of the use of "doped" fuel in the racing game and its present status. About 4 years ago Mr. Moskovics called up and asked us to send some dope over to the track. We sent some over by one of the men and said, "This may have some use in the racing game. If anyone would like to try it, we will try to work out the right proportions to give the best result."

Earl Cooper, Tommy Milton and Jimmy Murphy used it sporadically during the following season. They arrived at the fairly definite conclusion that it was helping them. In some cases, they got a little more speed out of the engine, but their general impression was that they were saving the engine. In coming out of the turns, they felt that the engines were not being stressed so hard and that they were running better.

The following year we went to Indianapolis and decided to make a real effort to get the drivers to use it. That was 2 years ago. About one-half the cars in the race used it. The following season Jimmy Murphy used it regularly and made his wonderful record. In last year's race about one-half the cars used it. DePaolo adopted it for his season last year. These facts seemed to convince the drivers that the fluid was "pretty good stuff" to put into their fuel. This year I think nearly every car that started in the race used ethyl gasoline. The main thing that any of these anti-knock materials does is to slow down the rate of combustion. If the combustion is not too rapid, no benefit will be derived from using an anti-knock material because there is no knock. When you are seeking extremely high performance and extremely high compression, you get combustions that are too rapid and you must slow them down to get the work out.

EFFECT OF USE OF TETRAETHYL LEAD

Having put in the proper amount of antiknock material so that detonation is not developed in the cylinder, to add still more fluid will slow down the normal rate of combustion so that the speed will decrease. The speed will increase until you have the right amount and will then decrease. That seemed to be hard to explain to the drivers. At first, they were afraid of it; it was not hard to keep them from using too much. Now, they have an idea that it is pretty good. We try to get them to use 5 or 6 cc. of fluid per gal., hand them some fluid and find later that they are using from 30 to 35 cc. I heard of one stranger who used 40 cc. per gal. in the race. He did not go very far.

[38] M.S.A.E.—Vice-president and general manager, Ethyl Gasoline Corporation, New York City.

In the use of tetraethyl lead there is formed a lead deposit that has probably been heard of as much as ethyl fluid. This is deposited on the spark-plugs. The spark-plugs show that the porcelain is not so good an insulator as it was before. That would be harmful if it were not for the fact that, in using tetraethyl lead in the fuel, the temperature of the spark-plug is kept down so that, in actual operation, the insulation of the plug is just as good as it was before; but you cannot use two or three times as much fluid, get two or three times as much deposit and expect the spark-plugs to continue to function.

With regard to super fuels, such as are needed in the Indianapolis race, I do not wish to convey the impression that any gasoline, by adding to it some ethyl fluid, can be made good enough to put into these cars. We start with about the best fuel that nature makes. So you see we are starting well up from the standpoint of antiknock materials to begin with. When tetraethyl lead is added, you can run these 5.3 plus 16 lb. in the manifold, and the engines operate with it pretty well.

CHAIRMAN MOSKOVICS:—While Mr. Midgley was talking, Mr. Miller said that they tried to run without tetraethyl lead and they were not successful at all; the use of it was so remarkable that none of the engines could go without it. They put in from 10 to 40 cc. for 5 gal.

MR. OLDFIELD:—Mr. Harley spoke about the British being pretty good engine-builders. I have heard some remarkable stories about the Schmidt 21-cu. in. engine. I wonder whether competition against British engines has indicated that they are better than those we have been able to build of that size. What are the volume and the velocity of the air required to cool the particular 21-cu. in. engine that Mr. Harley is experimenting with?

MR. HARLEY:—The A. J. S. Co. in England has developed a line of racing machines in the last few years and is perhaps the most successful company there. A few other special machines with overhead camshafts have made remarkable records. We make no tests on air at all. In working up these engines, we have blocks that give us as close results as those we get on the track. We cool the engines from the front with a blast that is approximately the same as that obtained under actual speed conditions. No measure of air velocities that I know of has been made.

MR. WALL:—The racing game as it used to be and as it is today has been very fascinating. Beginning with the Vanderbilt Cup races, I know that all who saw those early races must have got a great thrill. At that time it looked very much as if it would take American manufacturers a long time to get to the point where they could compete with foreign racing-cars. We even stood in awe of the foreign racing-drivers. It was very much like the American troops and the German well-trained troops that we met in the war, until we learned to know them better.

The American driver would probably have developed faster if the designs of American racing-cars had developed a little faster.

EARLY VANDERBILT CUP RACERS

The early Vanderbilt Cup racers were large heavy cars. The speeds in the earlier races reached possibly 90 m.p.h., which at that time seemed twice as fast as those of later cars, which run 130 to 140 m.p.h. It was not until 1910, I think, that American designers really began to take hold of the racing-car problem. In the meantime, of course, a number of American racing-cars had been designed. That is when Mr. Duesenberg started. Let me add that

Mr. Moskovics was one of our early racing-car designers.

At the Grand Prix race, the American cars withdrew. It was rather discouraging at that time and I wondered when American designers would ever build cars to compete with the foreign creations. It was wonderful to see some of them. Drivers like Lancia and others added a great deal of zest to sport of that kind.

In 1910, the American manufacturer began to see whether he could compete. Up to the time of the war he had accomplished much. We won most of our races in the meanwhile, between 1910 and 1917, though a couple of times foreign cars had come over and won the Indianapolis Speedway race.

The cars had been going down in weight, the piston displacement going up in speed. Now we have a car, due principally to Mr. Miller and Mr. Duesenberg, that is vastly different from our previous racing-cars.

Principal Troubles of Racing-Cars

As Chairman of the Technical Committee, I have had to do principally with the troubles that develop on these cars. In the early days it was such things as axles, steering-knuckles, wheels, springs, and things of that kind that gave the most trouble. Today, it is the minor details. In the last race, the troubles that were experienced were not of the nature of those which we had to a great extent in the early days, but were due principally, I think, to the high speed of the engines and to details such as valve springs, and, in the case of one of the foreign cars, lubrication troubles and details that the American manufacturer today is looking after a thousand times better than our foreign competitors, though I wish to give all credit to those who have come over here from abroad and put their cars into these races.

We have experienced very little trouble in the breaking of steering-knuckles in the last few years. One car that was entered in this last race had a steering-arm break. It was a foreign car.

Influence of Racing on Design

The effect of racing on design and manufacture has been great. A great many persons say that they cannot see how the racing business has helped the builder of passenger cars, but I think it has had a great influence. It certainly has developed a high-speed engine. Although we are not using engines in passenger-cars up to the speeds of these later creations, the engine-speed of passenger-cars today is from 1000 to 2000 r.p.m. higher than it was 12 or 15 years ago.

One of the great things, however, that I think racing, on the Indianapolis Speedway especially, has done for the motorcar business is the development and heat-treatment of steels. Our early troubles were due mainly to the fact that we did not understand alloy steel; we did not understand proper heat-treatment. Before some of the Indianapolis Speedway races, it has been customary for many manufacturers to change both the front and the rear axles, if they did not start out fresh, so as to be sure that the axles would go through 500 miles without breaking. In some respects, that seems absurd.

At the same time, those who know the characteristics of the Indianapolis Speedway realize that the vibrations there, especially at speeds of 100 m.p.h., are so rapid that they will fatigue almost any ordinary steel unless it has had proper heat-treatment. In fact, even in passenger-car tests, I have seen cars running at only 50 m.p.h. on the Indianapolis Speedway break some springs after a few thousand miles. At 100 m.p.h., the vibrations are much greater.

The troubles experienced today with racing-cars are with minor details, small things that put them out of the race, in comparison with the troubles that they previously had.

CHAIRMAN MOSKOVICS:—Mr. Wall told you that what might be called the beginning of American supremacy in racing began in 1914; the car that he designed won the 500-mile race in that year.

Some of Mr. Miller's cars were magneto-equipped. No one has asked the relation between magneto and battery ignition.

MR. MILLER:—In testing out magneto and battery ignition, we have not found any difference in speed.

L. R. SMITH[27]:—Did most of the cars have magneto ignition, and why?

MR. MILLER:—One of the troubles we have with battery ignition is the battery. It is very hard to get a battery fully charged. The track is very rough. The magneto is a unit. In case of trouble, the trouble can be located and the whole unit can be changed in a very few minutes.

CHAIRMAN MOSKOVICS:—The drivers told me that the main thing against battery ignition was their inability to get intelligent service on special types of battery in different places.

MR. MILLER:—It takes a relatively long time to charge one of the little batteries. The rate of charging them is always crowded. Carrying a generator means carrying extra weight.

PROFESSOR HUEBOTTER:—Do you find that the finish of the outside of the cylinder makes much difference in the ability to cool the cylinder? Does a nickel-plated finish cool any better than a black finish?

MR. HARLEY:—We think a black finish is better than a nickel-plated finish. Nickel-plating is put on for appearance only. An ordinary rough black finish is better.

MR. OLDFIELD:—Do persons closely connected with racing think that there will be a revision of the rules that will penalize two-cycle engines? I believe the European rules provide for rating the two-stroke cycle at 1.6 as against 1.0 for the four-stroke. If there should be a reasonable development, would not the same penalization apply to supercharging?

CHAIRMAN MOSKOVICS:—I do not think there has been any such discussion up to the present time.

R. H. SHERRY[28]:—Mr. Wall brought up an interesting point on the subject of the materials in racing-cars. The specifications were about 450 Brinell on the steering-arms and steering-knuckles. That is probably one of the reasons for their breaking. Makers seem to think they must go the limit on steel, particularly alloy steel. As a result they end with a piece of hardened tool-steel.

They speak of crystallization. Of course, there is no such thing. There are fatigue and shock. So, two items must be considered in the racing-car and also in the passenger-car game; one of these is that the material must be hard enough to sustain the load; the other, that it must be soft enough to take care of shock. I wonder whether anyone knows just what the condition of that steering-arm was when it broke.

H. F. BRYAN[29]:—I investigated the steering-arm that broke at Indianapolis. It seemed that before the race they had used an oxyacetylene torch on both the steering-arms and had changed the angle of the arm where the tie-rod between the two arms was connected. They did that

[27] M.S.A.E.—Indianapolis.

[28] M.S.A.E.—Consulting metallurgical and industrial engineer, Evanston, Ill.

[29] M.S.A.E.—Assistant chief engineer, Ensign Carburetor Co., Chicago.

without taking the steering-arm out of the knuckle-joint and probably either quenched it or allowed it to cool in the air.

CHAIRMAN MOSKOVICS:—In other words, they did their own heat-treating locally.

F. E. BOOTH[30]:—Do front-wheel-drive cars require less gear-shifting when accelerating into the long straights?

CHAIRMAN MOSKOVICS:—There is no gear-shifting on the track.

MR. BOOTH:—I did not see the race. I assumed that the smaller engines in the lighter cars would require some gear-shifting. Was there any difference in the front-axle and the rear-axle-drive gear-ratios?

CHAIRMAN MOSKOVICS:—The two cars were of exactly the same weight and had exactly the same displacement, which makes that a rather nice point.

MR. MILLER:—The front-wheel-drive cars had a higher gear-ratio than the rear-wheel-drive cars because they do not have to slow up.

CHAIRMAN MOSKOVICS:—The front-wheel-drive cars came out from the turns immeasurably faster than any other car. They retained their speed all the way round. As a matter of fact, they did very little accelerating. It might interest you to know that, for physical reasons, Cooper lost three laps immediately beginning the race. The result was that he had to drive 4 or 5 sec. faster per lap than anyone else. Yet he could do that, and he passed most of the men on the turn because he had that advantage.

MR. HERRINGTON:—Most of the drivers of the rear-axle-drive cars went into the turn at about 7000 or 7200 r.p.m. and came out at about 5000 r.p.m. That is a drop of about 2000 r.p.m Can Mr. Miller give us the drop that took place on the front-wheel drive?

MR. MILLER:—The front-wheel-drive cars never had to go very fast at any point because they did not have to slow up; the speed was only from 6000 to 6200 r.p.m. Although other cars had greater speed, the drop in speed was greater.

CHAIRMAN MOSKOVICS:—The point is that they did not drop because of any power loss, but because they had to slow down at the turn, while the front-wheel-drive cars did not.

TIRE CHARACTERISTICS

PROFESSOR HUEBOTTER:—What is the difference between the balloon tires that have been used the last 2 years and the high-pressure tires that were used in the previous races?

MR. BRIGGS:—Present-day Firestones use only a four-ply cord on the balloons. They use very thin side-walls and very thin, very hard tread, on the Indianapolis track. The old ones were eight-ply.

[30] M.S.A.E.—Sales manager of motor bearings division, Hyatt Roller Bearing Co., Harrison, N. J.

[31] M.S.A.E.—Mechanical engineer in charge of engineering laboratory, Firestone Tire & Rubber Co., Akron, Ohio.

MR. HERRINGTON:—They were inflated to 100-lb. and the new ones to 35-lb.

MR. BRIGGS:—They carry only from 35 to 40 lb., depending on the temperature of the outside air. The tires must be very carefully balanced and have the same air-pressure.

MR. MANLY:—Can Mr. Briggs tell us anything about the maximum temperatures of the tires during the race? Did he have any impression either by feel or by measurement?

MR. BRIGGS:—You cannot touch them. They are very warm. They are boiling.

CHAIRMAN MOSKOVICS:—Practically every car in the race used Firestone tires, except the British, which used Dunlops.

R. W. BROWN[31]:—The temperatures developed, of course, are an indeterminate factor. They are functions of the wind velocity, of whether the sun is shining or it is about to rain, of how much the driver is stepping on the gas, and of how much he is sliding on the curves. Ordinary passenger-car tires driven on a gravel or a cement road develop temperatures ranging from 130 to 170 deg. fahr., that is, under continuous service conditions.

On drum testing-machines, we have found temperatures in excess of 250 deg. fahr. and, in some very aggravated cases, meaning overloads on the tires and low inflation-pressures, the temperatures rise considerably above the vulcanization point of the rubber, that is, from 270 to 310 deg. fahr.

It is possible to under-inflate or over-load, which are somewhat similar conditions in tires, and develop sufficient temperature between the tread and the carcass of the tire to melt or devulcanize the rubber.

Improvements in tire construction for racing-cars during the last year have been largely confined to developing resistance to abrasion of the tread surface by compounds and sacrificing other features of tire construction that are desirable for long wear in passenger-car service.

One difficulty that will probably persist, with conventional tire-construction, is maintaining the tread in intimate contact with the carcass. I believe that has been done more or less successfully this year. That feature, however, will certainly bear further development work, in that at speeds around 190 and 200 m.p.h. it is a serious consideration. This introduces the problem of reducing the weight of the tread to resist centrifugal action at the relatively higher number of revolutions that occurs in wheels of this diameter at speeds up to 200 m.p.h.

I predict that the development of racing-tires will probably be confined to these two phases of the subject for at least the forthcoming year, that is, the development of treads that will stand abrasion better and secure a much more intimate connection between the tread and the carcass.

Production of Gasoline Substitutes from Coal[1]

By A. C. Fieldner[2]

Chicago Section Paper *Illustrated with* Drawings and Photograph

ABSTRACT

No danger exists of the imminent exhaustion of the petroleum reserves of the United States, as is shown by a committee report published early in 1926 by the American Petroleum Institute, from which figures are given in the following paper. It is reasonable to assume that a sufficient supply of oil will be available for all purposes beyond the time when the demand therefor will be reduced by more efficient use of petroleum products or by the production of substitutes for them.

The possibility of a future shortage of petroleum fuel suitable for automotive engines, however, and of the production of substitutes to avoid such a contingency, is receiving considerable attention in America and Europe. The author presents a general review of the situation and the status of research in the manufacture of gasoline substitutes from coal, of which enormous quantities remain unmined in this Country. Four known methods of extracting such substitutes from coal are

(1) High-temperature carbonization of coal in by-product coke-ovens or in gas-retorts
(2) Low-temperature carbonization
(3) Hydrogenation of coal
(4) Synthesis of hydrogen and carbon monoxide gases derived from coal, resulting in the production of alcohols

Each method is described briefly and the physical and economic possibilities of each to augment the supply of motor fuel are discussed. Only the first method is an existing industrial process; the others are in the stages of development. The second and third methods seem to offer important possibilities in the relatively near future, while the synthetic process of producing alcohols from coal-gases is interesting from a theoretical point, as it indicates that hydrocarbons usable in present or slightly modified automotive engines can be produced at moderate cost from inferior coals.

Other products than motor fuels are produced by these processes, such as coke, heavy oils, and gases suitable for illumination and heating. The carbonization methods are dependent economically upon the sale of these in addition to the sale of the gasoline substitutes, but the hydrogenating and synthesizing processes may be self-supporting on the liquid products.

Coke produced by low-temperature carbonization is suitable for household use and may be pulverized and burned in industrial plants as powdered fuel; it is also suitable for use in gas-producers. Some motor vehicles are now operated in Europe on producer gas and it is likely that such a solid fuel will be used as a gasoline substitute in this Country in motor-trucks, motor-coaches and tractors that operate continuously during working hours.

Crude tar produced by low-temperature carbonization must be cracked to give any considerable yield of light hydrocarbon from coal, and the crude oil derived by the hydrogenating process is fractionated and distilled to give motor fuel, Diesel-engine oil, fuel oil, and gas.

The petroleum resources of the United States are reviewed and classified in a voluminous printed report submitted by a committee of 11 members of the Board of the American Petroleum Institute and published by the Institute early in 1926. The Committee estimates that 5,000,000,000 bbl. of petroleum in the ground in fields that are known is available by present flowing and pumping methods. An additional 25,000,000,000 bbl. will be left in the ground in the same fields after the 5,000,000,000 bbl. has been removed. The Committee makes no attempt to estimate the quantity of oil in the undiscovered fields of this Country or can it or anyone do so.

The Committee then discusses the reserves in oil-shale, which are estimated at 107,000,000,000 bbl. It also estimates that 545,000,000,000 bbl. of petroleum substitutes can be obtained by processing bituminous coal, such as exists in the Appalachian and Mid-Continent fields and in the West, and also sub-bituminous coal, which possesses properties between bituminous coal and lignite. Finally, 70,000,000,000 bbl. is estimated to be contained in the lignites of the West, principally in the Dakotas, eastern Montana, and Texas. These brown lignites are much leaner with respect to tars or tar-oils than bituminous coal. There is a total of about 722,000,000,000 bbl. of liquid petroleum in oil-shales and coal, compared with the known 25,000,000,000 or, say, 30,000,000,000 bbl. of free petroleum. While only 5,000,000,000 bbl. of the petroleum reserves is available by present pumping methods, a very large proportion, probably, of the 25,000,000,000 bbl. remaining in the oil-sands will be available in the future by improved methods developed for mining these sands.

Our coal resources are estimated by M. R. Campbell, of the United States Geological Survey, at about 3,500,000,000,000 tons. If all this coal, equivalent to 517 cu. miles, were spread over the State of Ohio it would cover the entire surface of 41,040 sq. miles to a depth of about 76 ft. Of this, only 14 miles square, equivalent to 2.8 cu. miles, or much less than 1 per cent, has been used up to the present. So, regardless of when it may be necessary to call upon coal for a gasoline substitute, we can be assured that a vast resource of coal is available.

Four Methods of Extracting Gasoline Substitutes from Coal

By what methods and how much of this coal can be converted into gasoline? Other motor fuels than gasoline can be considered, but for the present let us consider only a liquid fuel having characteristics relatively similar to the gasoline that is used today. Four possible methods of conversion are known, as follows:

(1) Carbonization of coal in by-product coke-ovens or in gas-retorts. This is a standard method now

[1] Published by permission of the Director of the Bureau of Mines.

[2] Chief chemist, Bureau of Mines, City of Washington; and superintendent, Pittsburgh Experiment Station, Pittsburgh.

in use and provides a fuel known as motor benzol. By this method from 3 to 4 gal. of light oil is obtained from each ton of coal carbonized

(2) Low-temperature carbonization. This is not yet a commercial process but is under experimental investigation on a large scale both in America and Europe, and the art is farther advanced in this Country than abroad. The process gives larger yields of a crude-petroleum substitute than the first process and similar yields of a light oil suitable for motor fuel

(3) Hydrogenation of coal by the Bergius method, whereby 1 ton of liquid fuel, petroleum-like in character, can be obtained from 2 to 3 tons of coal

(4) Synthetic methods of making volatile liquid fuels from gases, which, in turn, are made from coal. The products are fuels such as alcohols and mixtures of alcohols and hydrocarbons, called "synthol"

Of these four methods, the first, or high-temperature carbonization, is the only one that is an existing industrial process. It is today providing motor fuel that is used in cities near the coke-ovens. The benzol is especially desirable for automotive-engine use because of its antiknock characteristic. As a rule it is mixed with gasoline. The following review should indicate to what extent we may count upon these different methods and how much motor fuel they will provide.

Inadequate Supply from High-Temperature Carbonization

From 2 to 3 gal. of a refined motor fuel can be obtained from each ton of coal by high-temperature carbonization (1). The total quantity of coal coked in the entire Country, including coking in beehive ovens, is about 60,000,000 tons annually. If all the potential motor benzol is recovered from this coal, roughly 150,000,000 gal. will be obtained. This quantity is 1½ per cent of the gasoline consumption of 1925, which was approximately 10,000,000,000 gal. If all the bituminous coal that is mined, about 520,000,000 tons per annum, were treated in this way, it would give 14 to 15 per cent of the Country's total requirement. Obviously, if we are looking far into the future, other methods than the present high-temperature coking process of obtaining gasoline substitutes from coal must be developed.

It may be stated in passing that a method is used at Bethune, France, for recovering ethyl alcohol from coke-oven gas and gives from 1 to 2 gal. per ton of coal carbonized. This ethyl alcohol is made from the ethylene in the gas. The plant is an experimental one but indicates that it is theoretically possible to obtain an additional 1 per cent of our requirement by way of the ethylene from coke-oven gas.

Possibilities from Low-Temperature Carbonization

In the high-temperature carbonization process coal is heated to about 1800 to 2000 deg. fahr. From 10 to 12 gal. of tar and from 3 to 4 gal. of crude light oil are obtained per ton of coal carbonized. If this coal is carbonized at about half this temperature, that is from 1100 to 1200 deg. fahr., about three times as much tar, or 20 to 30 gal., is obtained from a ton of coal. This tar has different characteristics than the tar obtained by high-temperature carbonization. It resembles petroleum in many respects. One-third to one-half consists of tar acids and higher phenols and the rest is hydrocarbons, both saturated and unsaturated. It does not contain naphthalene nor anthracene, and very little phenol or cresol, all of which are found in high-temperature tar, but it does contain hydrocarbons similar to those found in petroleum. It has less heating value than petroleum, that is, 16,000 to 16,500 B.t.u. compared with 19,000 to

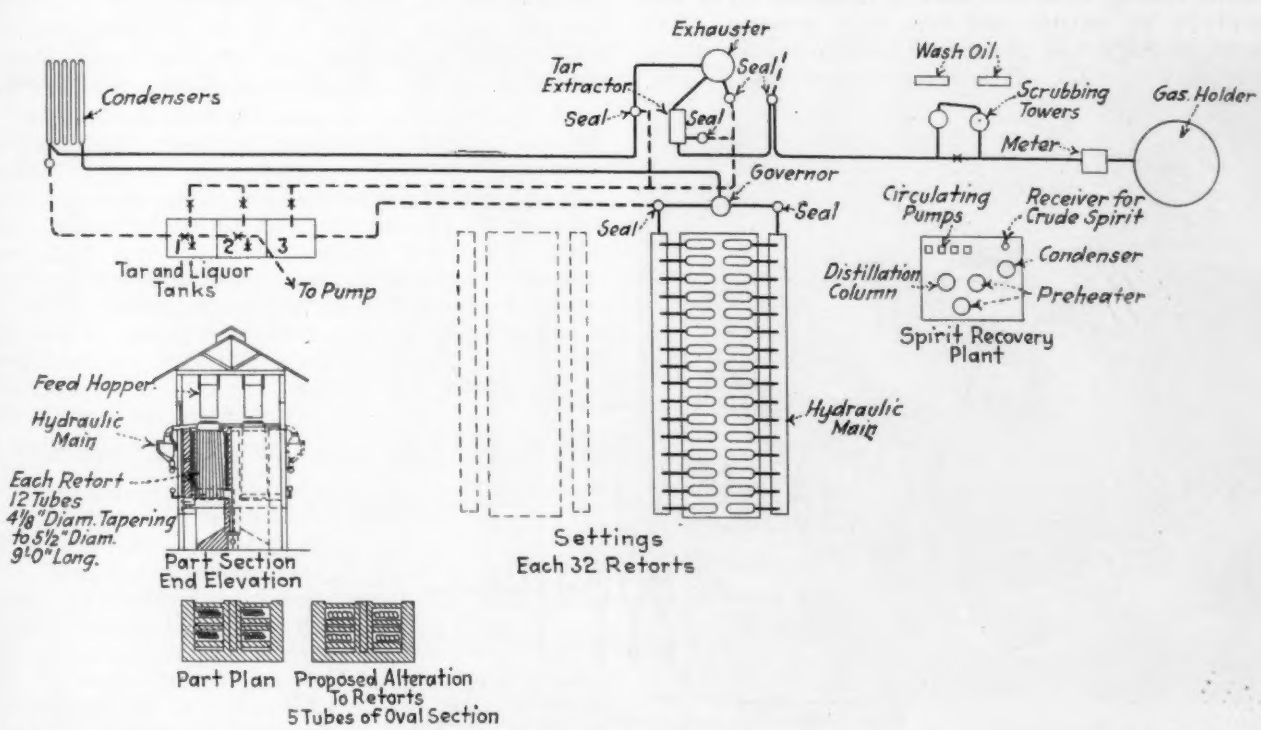


Fig. 1—Diagram and Elevation of Installation of Parker Low-Temperature Carbonization Retorts at Barnsley, England

This Installation Consists of 64 Retorts Having a Daily Capacity of 50 Tons of Coal. The Retorts Are Heated Externally to a Temperature of between 1100 and 1200 Deg. Fahr. and Charged Intermittently. The Carbonization Period Is 4 Hr. From 20 to 30 Gal. of Tar Is Obtained per Ton of Coal, and from One-Half to Two-Thirds of the Tar Consists of Hydrocarbons. From 1 to 2 Gal. of Light Oil Suitable for Internal-Combustion-Engine Fuel Has Been Obtained by Scrubbing the Gas and an Additional Gallon by Distilling the Tar. To Obtain a Larger Quantity of Gasoline Substitute, the Tar Must Be Cracked or Hydrogenated. Roughly 4½ Gal. of Light Oil Has Been Recovered from the Tar Experimentally by Cracking

20,000 B.t.u. per lb. of petroleum. From 1 to 2 gal. of light oil can be scrubbed from the gas and another gallon can be distilled from the tar, the total yield being from 2 to 3 gal. of light oil, which is roughly about the same quantity as is obtained from high-temperature carbonization.

This light oil, however, is subject to greater refining loss than benzol, so the net yield of light oils from most low-temperature processes is somewhat less than the yield of motor benzol from high-temperature processes. Therefore, to recover more gasoline substitutes by low-temperature carbonization it is necessary to crack the crude tar or subject it to some hydrogenation process. Experiments have been made whereby 18 to 20 per cent, by weight, of low-temperature tar has been converted into light oil suitable for automotive engines, giving roughly 4½ gal. per ton of coal. If this quantity can be obtained by a cracking process and, say, 2 gal. by direct recovery from the gas, we might look forward to a total of 6 gal. of motor fuel per ton of coal carbonized by low-temperature processes.

The question then arises, Why should we carbonize coal at a low temperature for only 6 gal. of motor fuel? Obviously, profitable operation of a low-temperature process must depend upon the combined yield of liquid, gaseous and solid fuel.

How the Coke and Gas Can Be Utilized

Low-temperature carbonization of coal yields a coke of different characteristics than the coke produced by high-temperature carbonization. The product of most processes, although not of all, is friable, weak and porous. It is lighter than by-product-oven coke or gas-coke and is not suitable for metallurgical purposes. It has a low ignition-temperature and is very reactive with air. It is easily ignited in an ordinary grate or stove, and burns readily with a smokeless flame, as all of the smoke-producing constituents have been removed. The coke contains from 6 to 15-per cent volatile matter which tends to easy ignition. If low-temperature coke is produced under pressure while in the plastic stage, the result is an ideal domestic fuel which would come in the anthracite class of fuels and probably would become popular with the householder. It is also possible to briquet soft low-temperature coke, recarbonize the briquets and produce an excellent domestic fuel.

Another possibility for deriving oil and gasoline substitutes from low-temperature carbonization is to combine the process with large central-station powerplants. Low-temperature coke is easily pulverized, hence lends itself to firing as powdered fuel. It is probable that processes will be developed in the near future whereby coal will be processed and the oils and vapors removed before it is fired under the boilers in large powerplants. General adoption of such treatment depends upon the market for the low-temperature tar, which is unfavorable at present due to the cheapness of oil. The supply of petroleum must be reduced considerably before market demands will permit the pre-treating of coal for powerplants at a profit.

The gas from low-temperature carbonization has a higher heating-value than coke-oven gas; 800 to 1000 B.t.u. as compared with 500 to 600 B.t.u. per cu. ft. About 3000 to 4000 cu. ft. per ton of coal carbonized is obtained in comparison with 10,000 to 11,000 cu. ft. of gas from by-product coke-ovens. Therefore there are possibilities of combining the processing of coal with the making of liquid fuel and a gas high in B.t.u. that may be used in certain localities for enriching purposes to increase the calorific value of gas provided by the water-gas method.

It is seen from the foregoing that low-temperature carbonization is not self-supporting as a method of providing gasoline substitutes. It must be combined economically with other industries, and even if it is estimated that a smokeless-fuel industry of 100,000,000 tons per annum would be developed, which is greater than the present output of 90,000,000 tons per annum, and that 60,000,000 tons would go to powerplants in such combination units, the total of these 160,000,000 tons

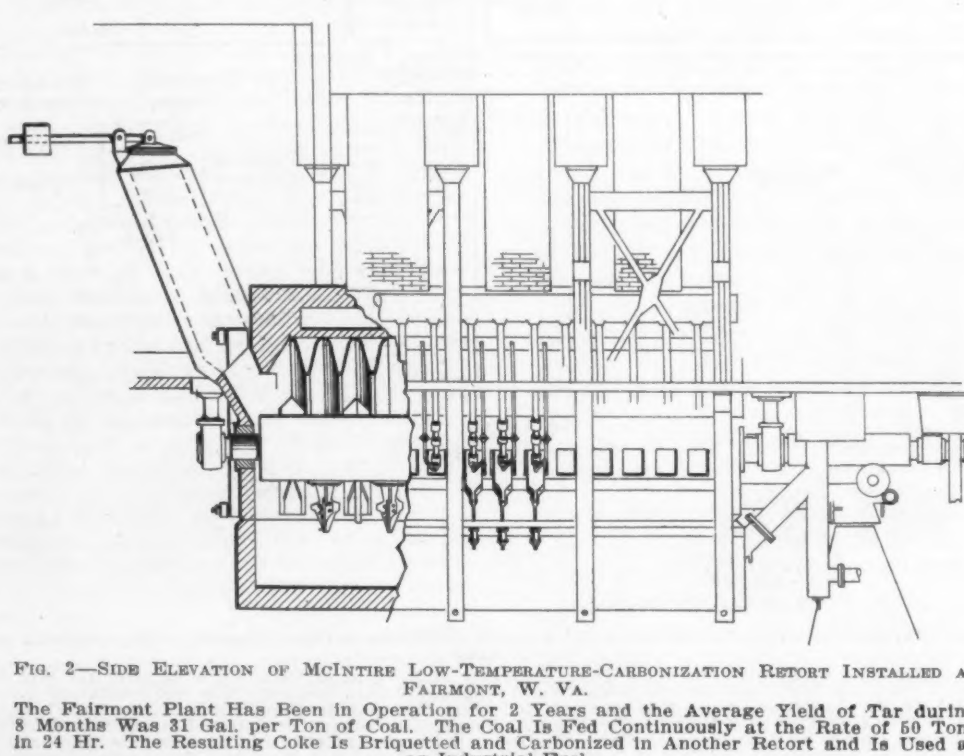


Fig. 2—Side Elevation of McIntire Low-Temperature-Carbonization Retort Installed at Fairmont, W. Va.
The Fairmont Plant Has Been in Operation for 2 Years and the Average Yield of Tar during 8 Months Was 31 Gal. per Ton of Coal. The Coal Is Fed Continuously at the Rate of 50 Tons in 24 Hr. The Resulting Coke Is Briquetted and Carbonized in Another Retort and Is Used as an Industrial Fuel

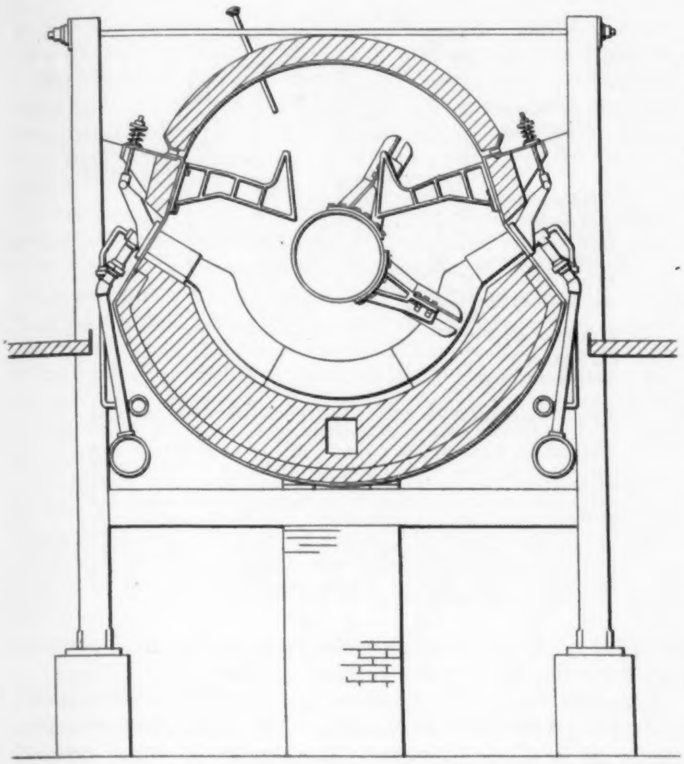


FIG. 3—END ELEVATION OF MCINTIRE RETORT
This Retort Is 16 Ft. Long and 6 Ft. in Diameter. It Is Heated Externally by Gas and the Coal Is Agitated by Stirrer-Arms inside the Retort

would provide only 9.6 per cent of present gasoline requirements. Nevertheless, this is a substantial figure in augmenting motor-fuel supplies.

Another phase of low-temperature carbonization recently came to my attention. The coke produced by these methods is very reactive and is a splendid gas-producer fuel; it does not melt in the fuel-bed, reacts readily with oxygen, leaves no unburned carbon in the ashes, and in every way is ideal for this purpose. One firm is marketing a domestic gas-producer unit for isolated country homes and for hotels at summer resorts or similar places. A very small and simple producer-unit provides the gas. Although designed to operate on charcoal, it operates well on certain low-temperature cokes. So, in the future, when petroleum is exhausted, we might use such gas-producers on trucks, motorcoaches and tractors. Some vehicles are operating on producer gas in Europe, and it is likely that the solid fuel from low-temperature carbonization may be used also as a gasoline substitute for certain types of motor vehicle.

SOME EXISTING LOW-TEMPERATURE CARBONIZING PLANTS

An installation of 64 Parker retorts[3] of 50-tons daily capacity at Barnsley, England, is shown in Fig. 1. The retorts are externally heated and are charged intermittently. The carbonization period is 4 hr. The coke produced is suitable for domestic fuel. The only objection to the process is that the costs are too high.

Figs. 2 and 3 show the McIntire retort, a development of the Carbocoal retort, operated by the Consolidation Coal Products Co. at Fairmont, W. Va. It consists of a horizontal retort 16 ft. long and 6 ft. in diameter. Coal is fed in continuously at the rate of 50 tons in 24 hr. The plant has been in operation for 2 years and the average yield of tar during 8 months is 31 gal. per ton of coal carbonized. The coke is briquetted and carbonized in another retort.

Fig. 4 shows the Thyssen system under construction at a colliery in the Ruhr region in Germany. The retorts are externally heated and produce gas of high calorific-value and a friable voluminous semi-coke which can be utilized only by briquetting with pitch or by pulverizing for powdered fuel. The retorts are 75½ ft. long and 8½ ft. in diameter and have a daily capacity of 100 tons of coal. Owing to the unusually high content of ethylene, propylene, propane and other easily condensable gases in the low-temperature gas, the Thyssen company has installed a Linde liquefaction apparatus for the separate recovery of these constituents and the light oils.

The approximate yields of high-temperature processes and the two main types of low-temperature processes, namely, externally and internally heated retorts, are given in Table 1. An analysis of these yields shows that the only increased by-product return is the additional 15 gal. of tar-oil per ton of coal carbonized over the quantity now being recovered in standard by-product ovens. As the price of this tar is determined by the prevailing price of fuel oil, which is about 5 cents per gal., the additional credit is only 75 cents per ton of coal. Against this must be charged a lower return of approximately 35 cents per ton of coal and a lower yield of gas. Obviously, the by-products of low-temperature carbonization do not, today, afford any greater financial return, if as great, as existing methods of high-temperature carbonization.

OIL RECOVERY BY HYDROGENATION

The hydrogenation of coal by the Bergius process is purely experimental but provides a means whereby from 2 to 3 tons of coal can be converted into 1 ton of a

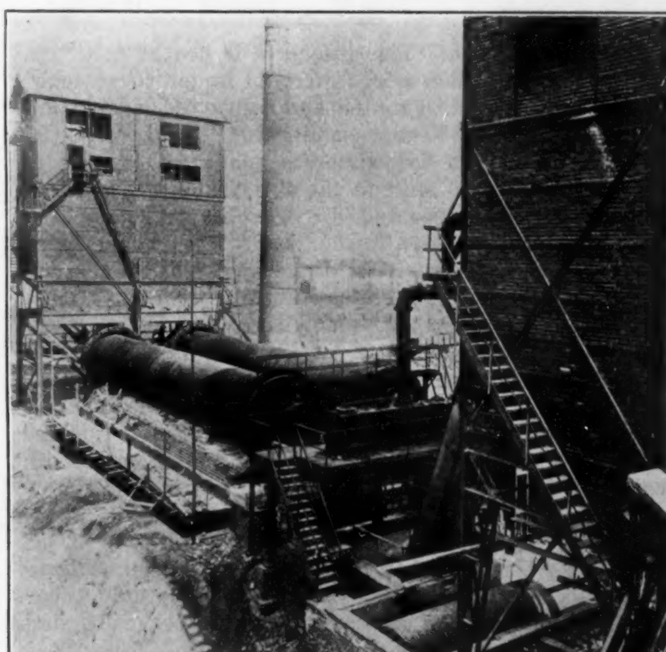


FIG. 4—THYSSEN DOUBLE-ROTARY LOW-TEMPERATURE CARBONIZATION-RETORTS UNDER CONSTRUCTION
This System Was Installed at a Colliery in the Ruhr District in Germany. The Retorts Are Heated Externally, Are 75½ Ft. Long and 8½ Ft. in Diameter and Have a Daily Capacity of 100 Tons of Coal. They Produce Gas of High Calorific-Value and a Friable Semi-Coke Suitable for Briquetting or Pulverizing. Liquefaction Apparatus Has Been Installed for the Separate Recovery of Light Oils and Easily Condensable Hydrocarbons from the Gas

[3] For complete details of this and the installations subsequently mentioned and illustrated see Bureau of Mines report, Fuel in Science and Practice, May, 1926, p. 203, and June, 1926, p. 265; also Bureau of Mines Technical Paper No. 396.

petroleum substitute. With a coal of optimum properties, there is only 15 to 20 per cent of solid residue, which consists of ash and some carbon. This method seems very attractive, because it can operate on low-grade coals; in fact, it works best with non-coking coals that are not suitable for making metallurgical coke. It also operates well with coals that are high in ash, of which there are great quantities in our Western States. A plant of this type could be operated anywhere in the West and would not depend upon any other industries absorbing some of the products, because virtually all of the solid coal can, theoretically, be converted into a petroleum substitute.

This process consists in mixing pulverized coal with oil or coal-tar to a thick paste which is pumped under a pressure of about 150 atmospheres into a steel autoclave

TABLE 1—APPROXIMATE COMPARATIVE YIELDS OF PRODUCTS FROM CARBONIZATION OF COAL BY HIGH-TEMPERATURE AND LOW-TEMPERATURE PROCESSES

	High-Temperature Carbonization in By-Product Coke-Oven	Low-Temperature Carbonization in Externally Heated Retort	Low-Temperature Carbonization in Internally Heated Retort
Coke, per cent	60 to 70	70 to 80	60 to 75
Volatile Matter in Coke, per cent	1 to 2	7 to 15	7 to 15
Gas, per Ton of Coal, cu. ft.	11,000 to 12,000	3,000 to 5,000	20,000 to 50,000
Calorific Value of Gas per Cubic Foot, B.t.u.	520 to 580	800 to 1,000	150 to 250
Tar, per Ton of Coal, gal.	10 to 12	20 to 28	18 to 20
Specific Gravity of Tar	1.19	1.07 to 1.09	1.02 to 1.07
Light Oil for Motor Fuel, gal.	2.5 to 3.0	2.5 to 3.0	None
Ammonium Sulphate, lb.	25 to 30	10 to 12	12 to 18

containing hydrogen. The autoclave is heated to a temperature of 750 to 800 deg. fahr. and under these conditions the coal is hydrogenated and converted largely into liquid compounds. Frederick Bergius, the inventor of the process, has an experimental plant at Mannheim, Germany, which was built in the early days of the World War and was first operated for cracking petroleum. It would produce from low-grade crude oils rather large yields of gasoline consisting of saturated compounds. The inventor has been experimenting with coal since 1918 and has operated a retort with a capacity of about 2 tons in 24 hr. Recently a larger unit was finished to handle about 40 tons in 24 hr. The yields are given as about 140 gal. of crude oil from a ton of gas-coal. This crude oil, when fractionated and distilled, yielded 40 gal. of a motor fuel, 50 gal. of a Diesel-engine oil, 35 gal. of fuel oil, a pitch residue, and about 10,000 cu. ft. of gas.

The retort requires about 5 per cent, by weight, of hydrogen per ton of coal treated. The solid residue from brown coals or lignite coals is reported to be only 1 per cent; from high-grade gas-coal about 10 per cent, and from a medium-volatile coal containing 20 to 25 per cent volatile matter, about 15 per cent of ash and undecomposed residue. The gasoline consists principally of aliphatic aromatic and hydroaromatic hydrocarbons; olefines are said to be absent. Therefore, the refining loss is claimed to be very low.

The hydrogen need not be pure; a purity of about 80 per cent is sufficient. The crude oil, or tar, if it may

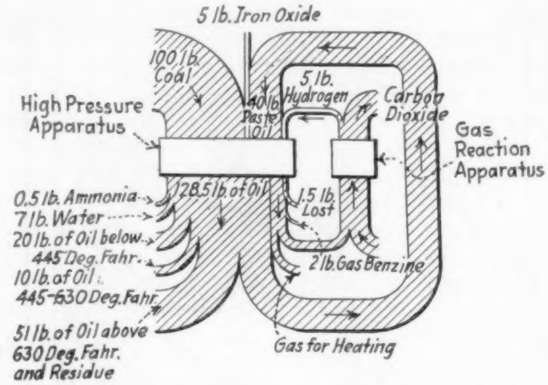


FIG. 5—DIAGRAM OF BERGIUS HYDROGENATION PROCESS FOR THE LIQUEFACTION OF COAL

Pulverized Low-Grade Coal Is Mixed to a Paste with Oil or Coal-Tar and Pumped under a Pressure of about 150 Atmospheres into a Steel Autoclave Containing Hydrogen, where It Is Heated to between 750 and 800 Deg. Fahr. Under These Conditions the Coal Is Converted Largely into Liquid Compounds and Yields about 140 Gal. of Crude Oil per Ton of Coal. When Fractionated and Distilled This Oil Gives 40 Gal. of Motor Fuel, 50 Gal. of Diesel-Engine Oil, 35 Gal. of Fuel Oil, and about 10,000 Cu. Ft. of Gas

be called such, contains phenolic compounds, the quantity varying with the oxygen content in the coal.

A recent report by Bergius gives results obtained on more than 100 European coals. The Coal Research Institute at Breslau, Germany, has conducted some experiments in hydrogenating tar by this process. When ordinary high-temperature tar from which all of the distillation products had been removed up to 170 deg. cent. (338 deg. fahr.) was treated, about 20 per cent more of materials that would distill up to this temperature was obtained. These consisted largely of aromatic compounds, benzene and toluene.

Fig. 5 is a diagram of the Bergius process for the liquefaction of coal. The yields from 1000 lb. of bituminous coal are given in Table 2.

The principal problems yet to be solved in the Bergius process are metallurgical and chemical; namely, the development of autoclaves that will withstand the high temperatures and pressures required and a cheap method of making hydrogen from the distillation gases. The method seems very attractive because it is self-sustaining and can be operated without depending upon the sale of the other products.

ALCOHOLS PRODUCED FROM GASES BY SYNTHESIS

The synthesizing of alcohols from gases has created a great deal of interest during the last year because of the importation of wood alcohol, or methanol, into America from Germany, where methanol is made by pressure synthesis from hydrogen and carbon monoxide. Two volumes of hydrogen and one volume of carbon monoxide are heated in an autoclave, or pressure-tube,

TABLE 2—YIELD OF PRODUCTS FROM 1000 LB. OF GAS-FLAME BITUMINOUS COAL CONTAINING 6 PER CENT OF ASH, PLUS 50 LB. OF HYDROGEN AND 50 LB. OF IRON OXIDE BY THE BERGIUS HYDROGENATION PROCESS

Products	Lb.
Motor Fuel	150
Fuel Oil for Internal-Combustion Engines	200
Lubricating-Oil	60
Fuel Oil for Industrial Use	80
Distillation Loss	35
Gas	235
Water	75
Ammonia	5
Coke, with Ash	240
Loss	20

to a temperature of 750 to 800 deg. fahr. under a pressure of 150 to 250 atmospheres and in the presence of a catalyst, said to be zinc oxide. The autoclave chamber must be lined with copper, as iron and nickel will prevent the reaction going in the direction of methanol only. It is said that methanol can be made for 13 to 20 cents a gallon in Germany.

This method also is self-sustaining, because water-gas could be made by standard methods and part of it converted with steam in the presence of a catalyst to hydrogen and carbon dioxide. The carbon dioxide could then be washed out and the carbon monoxide and hydrogen would be in the proper proportions to make methanol. A water-gas machine of a capacity of say 1,000,000 cu. ft. per day should make about 2500 gal. of methanol, of which from 1.6 to 2.0 gal. should be equivalent to 1 gal. of gasoline.

Unfortunately, methanol does not blend with gasoline, so this product, in itself, does not look promising as an augmenting agent for increasing motor fuel at a time when the bulk of the latter is gasoline. Obviously, a fuel is required that will enable a gradual change from gasoline to something else; it should be a fuel that will blend and that can be used in essentially the same engines as gasoline without any radical change in the engines. Nevertheless, methanol is interesting from the theoretical point in showing the possibility of making a motor fuel by synthesis from gases which are, in turn, made from coal.

In this connection Franz Fischer's work at Mulheim-Ruhr, Germany, is also of interest. He used a similar process in his laboratory but with a catalyst of iron impregnated with alkali, and obtained a mixture of substances which he called "synthol." This consists of organic acids, water-soluble alcohols and aldehydes, ketones and hydrocarbons. He obtained a yield of about 30 per cent from the gas that he started with; that is, 30 per cent in B. t. u. equivalents. This liquid has distillation characteristics similar to those of gasoline and a heating value of about two-thirds that of gasoline, and will operate internal-combustion engines. It will blend with both benzol and alcohol.

In the light of the foregoing review of the situation, no fear need be felt that if our supply of petroleum should be depleted rather suddenly a fuel shortage would result, because an abundant source of gasoline substitutes exists in coal. Furthermore, it is likely that much of the low-temperature tar will be used directly in Diesel-type engines, which would seem more economical than to crack it down to 6 or 7 gal. of volatile light oil. The Bergius process would provide, say, 40 gal. of gasoline from a ton of coal and an equal quantity of Diesel-engine oils. Possibly the residues can be retreated and more of the latter obtained. It is possible, also, that low-temperature coke will be used in gas-producer-equipped automotive vehicles, at least under conditions of continuous operation, as in motorcoach and freight-carrying services.

THE DISCUSSION

QUESTION:—In the low-temperature carbonization process are the boiling-points of the lighter oils changed and would they be higher or lower?

A. C. FIELDNER:—They cover a greater range; the oils begin to distill at 60 to 70 deg. cent. (140 to 158 deg. fahr.) and continue up to as high a temperature as one wants to carry the distillation. Low-temperature light oils consist of olefines, paraffines and hydroaromatics; they contain no benzol or toluol.

QUESTION:—Can they be used as motor fuel without being refined?

MR. FIELDNER:—No; refining is necessary and the losses by present methods are larger than for benzol because of the high unsaturation. Washing with acid and alkali entails a loss of approximately 40 per cent.

A MEMBER:—Can you tell us about the Salerni process that is being installed at the central electric plant in Brussels?

MR. FIELDNER:—The equipment is similar to the McIntire retort and consists of a horizontal retort with a device mounted on a central shaft for stirring the material and working it along the retort. A number of retorts are arranged in series. The plant operates only on non-coking coal.

A MEMBER:—I have been told that the investment was ridiculously low, about $300 per ton-day capacity, but I could not reconcile that with the cost of any process with which I am familiar. I thought the plant and process must be extremely simple. In this Country we estimate as low as $1,000 and as high as $5,000 per ton-day. I have seen no figure as low as $300.

QUESTION:—Do you think, in view of the present trend toward the development of special types of motor fuel, that the low-temperature coal-tar distillate will have something in its favor over, say, saturated Bergius distillate, because of the high aromatic and high olefine-content?

MR. FIELDNER:—The Bergius distillate is said to contain no unsaturates. It is true that research on anti-knock fuels has shown that unsaturates and aromatics in motor fuels permit the use of higher compression, hence are desirable constituents.

REFINING LOSS IS DUE TO UNSATURATES

A MEMBER:—I think the loss in refining a low-temperature coal-tar motor fuel, to which reference was made, is mainly in the tar acids rather than in the solids.

MR. FIELDNER:—The tar-acid content of the motor-fuel fraction is low. The tar acids are in the higher-boiling fractions and comprise more than 50 per cent of these fractions in some tars. Therefore the loss in the motor-fuel fraction when refining by the usual methods is principally due to unsaturates. I can imagine that these losses can be reduced by refining methods that are suitable for low-temperature motor fuel. Fischer published a description of a method of reducing tar acids to benzol with hydrogen in a tinned-iron pipe. We repeated his experiments in Pittsburgh and failed.

Obviously, if one can convert the tar acids into benzol the commercial value of low-temperature tar can be enhanced greatly, because from 30 to 50 per cent of the total tar-fraction consists of tar acids and acid resins.

DUST CLEANER NEEDED WITH PRODUCER GAS

R. E. WILSON[4]:—One point that possibly should be emphasized is that, in cracking the heavy tars, the coke production is so tremendous that types of apparatus rather different from those suitable for cracking petroleum probably will have to be employed.

Some question exists in my mind as to the promise of the gas-producer for automotive equipment. Since 1900 many patents have been issued on that general type of apparatus. One great difficulty with it always has been the entrainment of solid particles that cut the

[4] M.S.A.E.—Member of research council, Standard Oil Co. of Indiana, Whiting, Ind.

engine cylinders badly. Ordinary producer gas, carrying with it ash and unconsumed particles of carbon, has been reputed to give a great deal of trouble by wearing out the cylinders in a short time. If the gas is cooled and scrubbed this difficulty is eliminated but the complication is increased and the economy of operation is decreased. Gas-producers might possibly come into use for large units abroad, where gasoline is relatively expensive, but such a development here seems to me to be a long way off. Is the producer gas used hot in Europe or is it cooled before mixing with the air for burning?

MR. FIELDNER:—The gas-producers on automotive vehicles usually operate on wood charcoal, which has a very low ash-content and produces little tar, hence cleaning the gas is much simpler than it would be for coke. A unit I saw in England had a simple shaving scrubber. The gas was not water-cooled. The small domestic gas-producer outfit that I mentioned as now being placed on the market works well with charcoal and equally well with some low-temperature coke, but not with all. I should think that solid matter could be scrubbed out of the producer gas as well as dust can be removed from the intake air for the engine.

A MEMBER:—Some years ago a concern on the South Side in Chicago had a truck in which it had placed a gas-producer. As I recall, the gas reached the engine hot. The truck used to run to South Chicago and back using charcoal as fuel. I do not know that the owners ever tried any low-temperature-carbonized coke. I have understood that a process was tried at Pittsburgh of using steam in the neighborhood of 1000 to 1400 deg. fahr. to carbonize the coal at low temperature. Will you tell us something about that?

MR. FIELDNER:—Some experiments were made at the Bureau of Mines on a fellowship arrangement with the Carnegie Institute of Technology. Mr. Carrick carried on the work with Mr. Davis at the Bureau. It was a laboratory experiment to determine the effect of internal heating with superheated steam as compared with heating with products of combustion and of heating the coal externally. A proposal was made some years ago to carbonize coal by the sensible heat of superheated steam passed through the retort and circulating among the coal particles.

The retort in the experiments at Pittsburgh was a steel tube about 6 in. in diameter and 20 ft. high. The coal was fed through it continuously and steam, superheated to a temperature of about 1200 to 1400 deg. fahr., was introduced at the bottom. The tar and gas were collected and examined. The coal was a Utah non-coking coal. The fuel obtained by the process was a form of activated carbon. It ignited in a muffle that was heated just below a visible red. Carbonizing the coal in steam probably had something to do with the reactivity of the coke.

COTTON

ONE can but wonder if the farmer gives any thought to the effect upon his own fortunes or the fortunes of others when in the spring he plants 10 acres in cotton whereas he formerly planted 7. Whether he did or not, the result of his increased planting last spring, in the face of official and private warnings not to do so, makes it look as if he attempted to disregard history and defy the law of supply and demand. The recent shrinkage in cotton values is just another example of the many failures that have followed such attempts, and producers are now paying very dearly for their folly in a way that they are not apt to forget for at least a few years to come.

The four short crops from 1921 to 1924 brought high prices and stimulated production on the part of the producer of cotton the world over. As a result, the acreage of recent years especially in this Country has steadily increased from an average of about 36,000,000 acres to nearly 49,000,000 this season. The yield per acre has also increased of late owing to the greater use of fertilizers, improved methods of cultivation and decreased damage from weevil and other insects.

That past experiences would teach farmers not to attempt to raise two big crops in succession, especially with a heavy carry-over from the previous one staring them in the face, seems almost obvious. Statistics available in the spring pointed to a surplus or carry-over from the 1925 crop of between 5,500,000 and 6,000,000 bales. Final returns reduced it to 5,362,000.

In the past all big crops, except the one raised in 1925, brought low prices and were usually followed by a period of depression among farmers. The comparatively high price paid for the record-breaking crop of last year, though well under that received for the one grown in 1924, seem to have clouded the judgment of producers and to have been primarily responsible for their efforts to duplicate that record, if not to exceed it. At any rate they went ahead and planted nearly 1,000,000 acres more to cotton than was cultivated during the previous season. To understand the motive that prompted such action on the part of the producer in view of the fact that he always received more money for what he raised and was more prosperous during years of short or moderate crops is hard.

For the South to produce another big crop next year, and possibly for several years to come if history repeats itself, is an economic and financial impossibility. Production records for the last 50 years show that every big crop, with the exception of the one produced in 1925, has always been followed by one that was considerably smaller. Usually a series of small crops has followed. Low prices have accompanied all other big crops and producers were without the means or incentive to attempt to raise another large one. Large decreases in acreage have invariably followed low-priced crops. The record also shows that good prices were always received for the small succeeding crops. I believe that the final monetary returns for the crop will show that we are not hit as badly as we think and that the situation will correct itself far sooner than is now generally believed to be possible.—I. V. Shannon in *Trade Winds* (Union Trust Co. of Cleveland).

FOREIGN TRADE

IN the last 10 years our foreign trade has doubled. We have invested abroad, exclusive of the money owed to the Government, over $10,000,000,000, an increase of 400 per cent over the pre-war period. Our shipping has increased from about 750,000 to 11,000,000 tons and our tourists have increased from approximately 200,000 to considerably more than 500,000.—J. P. Atwood in *Trade Winds* (Union Trust Co. of Cleveland).

Vol. XX January, 1927 No. 1

Cushion Tires and Their Relation to the Vehicle-Operator's Tire Problem

By A. L. Schoff[1]

Transportation and Service Meeting Paper *Illustrated with* Drawings, Charts and Photographs

ABSTRACT

FACTORS that led to the development of solid, cushion and pneumatic tires are reviewed briefly, beginning with the substitution of solid-rubber tires for steel tires on light carriages and cushion tires for solid-rubber tires on bicycles. Because heavy motor-vehicles were insufficiently protected from road shock by solid tires, M. C. Overman developed a spring wheel in 1908 and later sought to improve its riding qualities by developing a cushion tire. After many years' work, a proper design was evolved and tests showed that the tire was so effective that the spring wheel was not needed. This type was sold extensively for use on heavy passenger-cars and light trucks, and development was continued to produce a suitable cushion tire for heavier trucks.

Because vulcanized rubber is non-compressible but is flexible and easily deformable in shape, a desirable cushion tire has an ample annular central cavity or its equivalent and a suitable tread shape to allow for the free displacement of the rubber and is designed so that no portion of the rubber will be overstressed when deformed, which would result in early failure. Qualities that such a tire should possess are enumerated.

Responsibility for the lack of satisfactory tire performances is laid largely upon the truck builder for failure to provide wheel felloes and felloe-bands sufficiently wide to give adequate support to the edges of the base of oversize tires, which frequently must be fitted. Because of this lack of support the overhanging base is bent by striking curbs and high car-tracks, and separation of the tire from its base results. Many States limit the gross weight that may be carried per inch width of tire, which makes it necessary for the truck operator to use wider tires than the original equipment, and the wheels are then undersize. The author suggests that the Standards Committee of the Society can do some constructive work toward lowering truck operating-costs by going more fully into the question of wheel standards and designs. As a rule a cushion tire for a heavy vehicle has a base that is 1 in. wider than that of a solid tire of equal load capacity to compensate for the internal cavity.

The greater thickness or height of tread rubber available for wear in the cushion tire, combined with the easy deformability of the rubber, results in greater mileage from cushion tires than from other types. Records taken from large fleets of trucks show that mileages of 50,000 to 60,000 are frequent and that the average for cushion tires is 30,000 miles, for solid tires 14,000 and for pneumatic tires 12,000. Although the initial cost of cushion tires is more than twice that of solid tires and not very different from that of pneumatic truck tires, the tire-mile cost is shown to be less.

Impact tests made by the Bureau of Public Roads indicate that the cushioning effect of cushion tires is nearer that of pneumatic than of solid tires. They are well adapted, also, to molding in special designs for special purposes, and a design has been produced that gives excellent service in the roadless oil-fields and on sandy roads where formerly only pneumatic tires could give the necessary traction and support for the load.

Cushion tires are asserted to be more economical than other types because they (*a*) do not occasion delays on the road due to tire trouble, (*b*) do not necessitate emergency replacements and therefore eliminate the need for carrying spare tires in stock and on the vehicle and for maintaining an emergency tire service, (*c*) are easily deformable and have more tread rubber and therefore wear longer, (*d*) have better traction and non-skid properties and therefore do not so often require the use of tire chains and reduce the number of accidents, which lowers the premiums on insurance, (*e*) cushion the vehicle better than solid tires and therefore greatly reduce the breakage of component parts of the truck and the need for carrying stocks of replacement parts, and (*f*) permit faster running speed than solid tires, which increases the mileage and the number of unit deliveries per truck per day.

Failures of cushion tires that occur usually are the result of improper selection of size, improper truck maintenance, overloading, or overspeeding. Five common types of failure are (*a*) rapid tread-wear on front wheels, (*b*) uneven tread-wear, (*c*) flat spots, (*d*) burn-outs, and (*e*) base separation. The principal contributing causes of each are given by the author, who also presents a table showing the usual types of tire equipment regarded as correct for vehicles of various types and capacities operated at different rates of speed.

IT may be of interest to review briefly some of the factors that have been responsible for the development of all kinds of rubber tire during a number of years to their present form and quality before dealing specifically with the cushion tire. The changes from steel-rimmed to solid-rubber tired light carriages and from solid-tired to pneumatic-tired bicycles and light motor-vehicles were made to lessen the shock to riders and vehicles from road irregularities. The first cushion tire developed was used on the bicycle to provide better riding-qualities than were afforded by the solid-rubber tire. This development resulted in the type of tire shown in Fig. 1, in which good cushioning properties were provided by molding the rubber so that it could deflect by bending under the load and thereby absorb within itself a large portion of an irregularity in the road as

Fig. 1—General Type of Bicycle Cushion-Tire
Tires of This Type Displaced the Solid-Rubber Tire and Were Used Extensively in the Early Nineties Prior to the Successful Development of the Pneumatic Tire. This Form Was of Reasonably Small Size and Was Suitable for the Light Weight Carried

[1] M.S.A.E.—Assistant vice-president, Overman Cushion Tire Co., New York City.

the wheel passed over an obstruction. A tire of reasonably small cross-section, when stressed in this way, performed satisfactorily on such light vehicles, and cushion tires of this general design were in common use in the early nineties.

Heavy vehicles were carried almost exclusively on solid tires at the beginning of the era of automotive transportation. Such equipment did not protect the vehicle from road shocks very effectively, however, and pneumatic tires made of fabric and rubber were entirely unsatisfactory in the sizes that were necessary for motor-trucks. Many attempts were made, therefore, to secure better riding-qualities by the use of cushion wheels fitted with solid tires. A spring wheel that was satisfactory in many respects was developed in 1908 by M. C. Overman, originator of the present Overman cushion tire, but to improve its riding qualities it was desirable to have a more flexible tire than the solid type. The development of such a tire was started but many years were spent in experiments before experience, added to theory, indicated the necessary design. When this was determined, tests of the tire showed that it was so satisfactory that the spring wheel was not needed. Extensive sales of the new type of tire were made for use on heavy passenger-cars

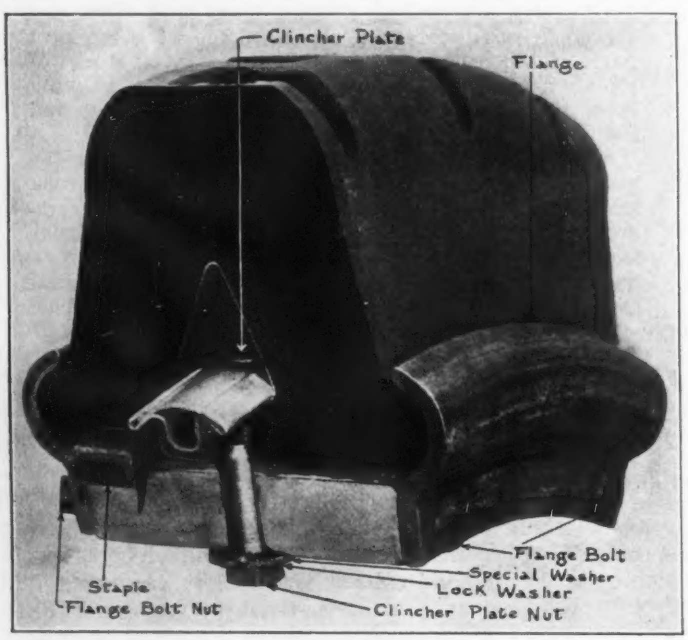


FIG. 2—EARLY TYPE OF CUSHION TIRE FOR MOTOR-VEHICLES
This Type Was Used Somewhat Extensively on Heavy Passenger-Cars and Light Motor-Trucks. It Was Not Vulcanized to the Base and Therefore Required a Special Rim, as Shown

and small motor-trucks, and development was continued steadily to produce a tire suitable for heavier trucks. All of these cushion tires were of a type that was not vulcanized on a steel base and which required a special wheel or rim for installation on the vehicle, as shown in Fig. 2.

WHY THE DESIGN IS IMPORTANT

Vulcanized rubber is practically non-compressible but is elastic and readily deformable in shape either by stretching, contracting or deflecting. A tire of the cushion type, to have the desired properties, should be designed with ample air space to allow for the free displacement of the rubber and so that no portion of the rubber, when deformed, will be overstressed, as excessive stress will result finally in failure at the point of overstress. Fig. 3 represents a tire that gives de-

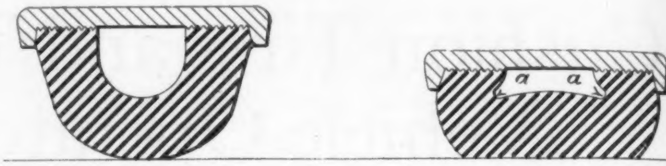


FIG. 3—FORM OF CUSHION TIRE THAT DEFLECTS MAINLY BY BENDING
When under Heavy Load, as at the Right, Excessive Bending Stresses Occur in the Areas *a a* and Lead to Early Failure of the Rubber at These Places

flection mainly by bending, which creates high stress in the areas *a a* and leads to early failure. It is most important that the design of the tire shall assure correct disposition of the rubber to avoid such areas of excessive local stress and that the design shall provide a proper profile with a suitable internal cavity, or its equivalent, and a suitable tread shape, as in Fig. 4, in which the whole cross-section of rubber undergoes substantially uniform compression without having any areas of high local stress even when the tire is subjected to excessive deflection due to overload or shock, as indicated at the right in the drawing.

Only two companies were making and marketing cushion tires for motor-vehicles in 1920 and their sales volume was comparatively small, but now at least nine prominent tire companies are marketing them. All of these makes of tire provide a varying amount of cushioning by more or less easy displacement of the rubber into an internal cavity and into tread-relief recesses. The increase in the number of cushion-tire makers has been accompanied by a large increase in the number of cushion tires sold.

DESIRABLE QUALITIES IN A CUSHION TIRE

A desirable cushion tire should have the following specific qualities, all of which effect to some extent the tire problem of the vehicle operator:

(1) Maximum cushioning consistent with large mileage
(2) Cushioning which is affected to the minimum extent by wear or aging of the tire
(3) Suitable tractive and non-skidding properties
(4) Safe lateral control
(5) Low cost per mile of operation
(6) Toughness to resist abrasion and cutting

An operator who expects to receive maximum performance from a tire of any type should be able to furnish accurate data on all of his operating conditions; only from such data can the most economical tire installation be worked out. Unfortunately for the entire industry, the truck builder is largely responsible, in many cases, for the lack of satisfactory tire performance. Even today too few trucks are produced which are designed so that the owner can fit the oversize tires which he finds are necessary without making expensive

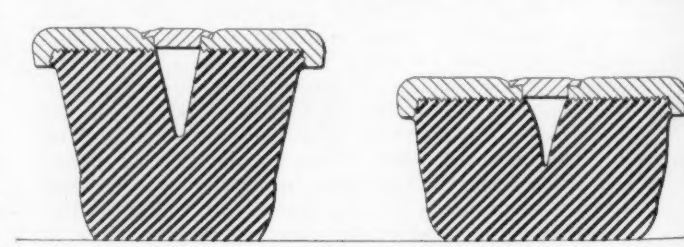


FIG. 4—PROPER DESIGN FOR CUSHION TIRE
Correct Disposition of the Rubber Permits Compressive Deformation of the Rubber into the Central Annular Cavity and to the Outer Sides without Bending and without Creating Areas of Excessive Local Stress. Because of the Tread Shape and the General Profile, the Whole Cross-Section Undergoes Substantially Uniform Compression, Even When the Tire Is Subjected to Excessive Deflection, as at the Right

alterations to either the wheels or the bodies or to both. This statement is equally true with regard to trucks that are originally equipped with pneumatic, solid or cushion tires. On how many vehicles equipped with pneumatic tires can oversize tires be substituted without encountering lack of clearance? On how many solid-tired vehicles can oversize tires be fitted without changing the wheels? The present narrow felloe-bands make it necessary to resort to single-flanging the tires and to use extension rings in an attempt to support the overhanging base of the larger tire, a most unsatisfactory makeshift. For example, the wheel equipment on most 5-ton trucks provides for either dual 40x6-in. or single 40x12-in. tires, a size that is entirely too light to carry the load that usually is put on a 5-ton truck. The two types of original tire equipment of 5-ton trucks are shown in Fig. 5, and the effects of putting proper oversize tires on the wheels with single flanges and with overhanging bases are shown in Fig. 6. The latter drawing shows the seriousness of the situation. The truck owner, after wearing out his first set of tires, is advised by the tire company to apply 40x7-in. dual tires or, in an extreme case, 40x8-

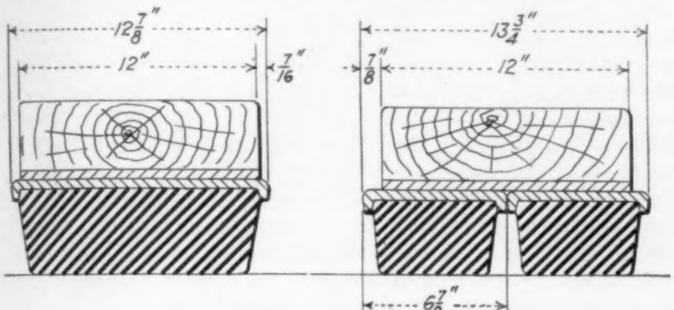


FIG. 5—ORIGINAL TIRE-EQUIPMENT OF 5-TON TRUCKS
Single 12-In. and Dual 6-In. Solid Tires Are Mounted on a 12-In. Felloe-Band. The Outer Edges of the Bases of the Dual Tires Overhang the Felloe-Band ⅞ In., or Double the Distance in the Case of the Single Tire

in. dual or 40x14-in. single tires. When this is done, the overhang of the tire base is excessive and the base is certain to be damaged by striking curbs or high car-tracks, often resulting in separation of the base from the tire and the early destruction of the tire. If the felloe-bands were made originally about $1\frac{1}{2}$ in. wider, the operator would save the loss of several hundred dollars' worth of tires. The seriousness of the bending of the tire base cannot be over-emphasized. Filler or extension rings, as shown at the left of the felloe-band in Fig. 6, are of only minor assistance, as in many cases they bend with the tire base. On certain types of wheel the felloe itself bends with the tire base, and such bending always leads to base separation.

I believe that the Standards Committee of the Society can and should do important constructive work toward

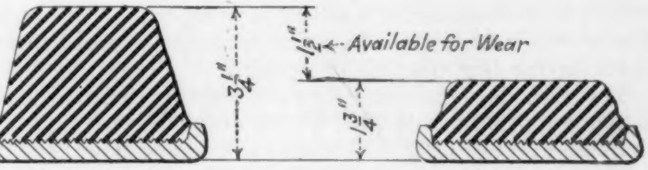


FIG. 7—TREAD RUBBER AVAILABLE FOR WEAR IN A 36X4-IN. SOLID TIRE OF 1700-LB. CARRYING CAPACITY
Rate of Wear Is, in General, Inversely Proportional to the Vertical Deformability of a Tire. This Tire, after Wearing Down 1½ In., Will Be So Worn and Chipped as To Require Replacement. It will Do Well If It Delivers from 15,000 to 18,000 Miles of Service

lowering truck operating-costs by going more thoroughly into the question of wheel standards and designs. Many States now require the operator to increase the size of tires to carry normal loads, and more States undoubtedly will do so. To comply with such laws or to obtain the best tire service, the operator must change his equipment and use a wheel that is in fact undersized, as a result of which he necessarily receives poor tire service. I suggest, therefore, that the operator, after determining the proper size of tire equipment, make sure that the truck builder supplies the proper wheel foundation for it. This may involve a few dollars of extra first cost but it will save many times that cost in better tire service.

Many persons have the erroneous idea that the manufacturer, when furnishing cushion tires, sells them tires that are unnecessarily large. It is necessary, as a rule, to use a cushion tire that is 1 in. wider at the base than a solid tire of equal carrying capacity to compensate for the width of the usual internal annular cavity in the body of the cushion tire, the chief function of which is to provide displacement space for the rubber in the legs of the tire. A 5-in. cushion tire, for example, is rated at and actually should carry only 1700-lb. load, which is the capacity of a 4-in. solid tire.

THICK TREAD RUBBER GIVES HIGH MILEAGE

Let us now review in some detail the more important features of cushion tires and some of the results obtained from them in service.

A suitable cushion tire will give the greatest number of days of uninterrupted service or the highest average ultimate mileage of any type of tire that is available today. Cushion-tire records of 50,000 to 60,000 miles per tire are frequent and the operator who maintains proper supervision over his equipment may reasonably expect an average of more than 30,000 miles per tire without the loss of any time due to tire changing. A large oil company placed ten 650-gal, tank-trucks fitted with cushion tires in service in the spring of 1923 and today, $3\frac{1}{2}$ years later, after the trucks have traveled an average of 35,000 miles, all but three of the original tires are still in service. Two tires were removed because they were worn out and the third was discarded after

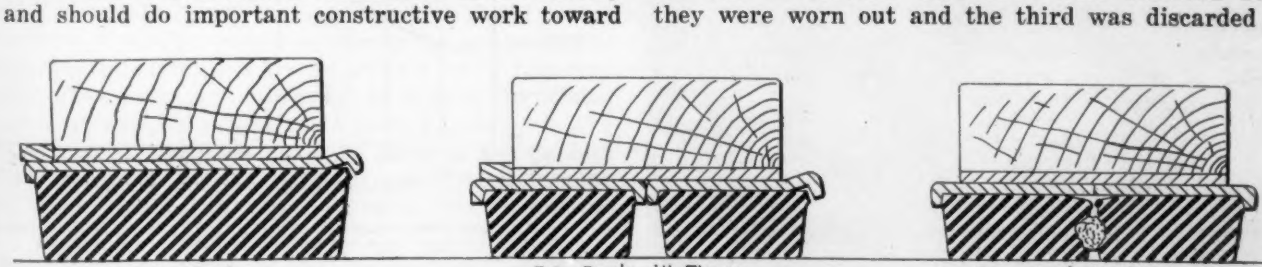


FIG. 6—EFFECTS OF FITTING PROPER OVERSIZE TIRES
When a 12-In. Felloe-Band is Fitted with a Single 14-In. Tire or Dual 7-In. Tire, the Tire Base Overhangs the Band an Excessive Distance and Is Certain To Be Bent, as Shown in the Left and Center Drawings, by Striking against Curbs and High Car-Tracks. If Dual 7-In. Tires with Single-Flange Bases Are Fitted, as in the Drawing at the Right, Small Stones Become Wedged between the Tires. In Either Case Separation of the Tire from Its Base Results. The Use of a Filler Ring or of Extension Rings, as Indicated at the Left of the Felloe-Band in the First and Second Drawings, Is of Little Avail, as in Many Cases They Bend with the Base

22,000 miles because of a defective weld in the tire base. These trucks have been in continuous use without interruptions for tire changes or repairs.

The records of another large operating company that is using approximately 1400 trucks of 2 to 5-tons' capacity show an average of 30,000 miles for cushion tires, 14,000 miles for solid tires and 12,000 miles for pneumatic tires. These figures indicate that cushion tires, initially costing more than twice the cost of solid tires, are an economical investment and the tire-mile cost as compared with that of pneumatic tires is very much less, since the initial cost of pneumatic and cushion tires is not widely different. The average mileages stated were obtained over a 4-year period, which is believed to be long enough to be conclusive.

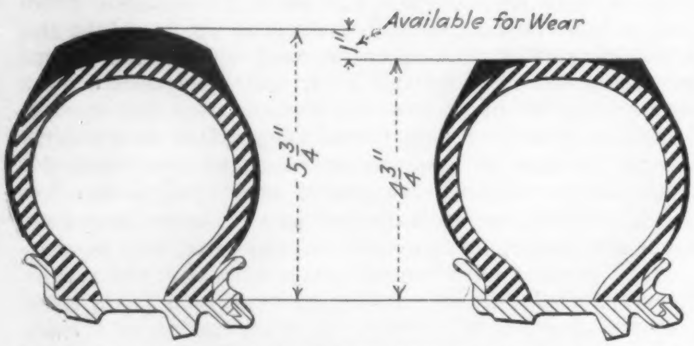


FIG. 8—TREAD RUBBER AVAILABLE FOR WEAR IN 34X5-IN. PNEUMATIC TRUCK-TIRE OF 1700-LB. LOAD CAPACITY

Owing to Its Easy Deformability, the Pneumatic Tire Will Run from 25,000 to 30,000 Miles Before the ¾ or 1 In. of Tread Rubber Wears Down to the Fabric, Provided no Failure of the Carcass Occurs

The rate of wear of a tire is, in general, inversely proportional to the vertical deformability of the tire. A solid tire that has available for wear at most a 3-in. height of rubber will do well if it delivers from 15,000 to 18,000 miles of service before it is so worn and chipped as to require replacement (See Fig. 7). A pneumatic tire that has 1 or 1¼ in. of tread rubber over its carcass usually will run from 25,000 to 30,000 miles before wearing down to the fabric, provided the carcass does not break down (See Fig. 8). Cushion tires occupy a place between solid and pneumatic tires in respect to vertical deformability. Most of them have from 1½ to 2½ in.

TABLE 1—RATES OF WEAR OF CUSHION AND SOLID TIRES, MEASURED IN ACTUAL SERVICE

Type of Tire	Size, In.	Average Miles Per ¼-In. Wear
Front Tires on 2 to 2½-Ton Trucks		
Cushion	36 x 5	6,400
Rear Tires on 2 to 2½-Ton Trucks		
Cushion	32 x 10	5,750
Cushion	36 x 9	5,900
Solid	36 x 8	4,440[a]
Front Tires on 3½ to 5-Ton Trucks		
Cushion	36 x 6	5,265
Cushion	36 x 7	6,900
Solid	36 x 6	5,475
Rear Tires on 3½-Ton Trucks		
Solid	40 x 10	3,516[a]
Solid	40 x 12	4,225[b]

[a] Average of four makes.
[b] Average of two makes.

of tread rubber available for wear varying with sectional width before they will split open to the internal cavity and will average from 24,000 to 50,000 miles, or from 4000 to 6000 miles per ¼ in. of wear, as is shown by numerous records of tires in every-day service on all kinds of truck (See Fig. 9).

The average rates of wear of tires as given in Table 1

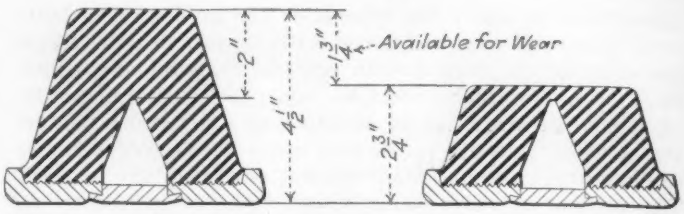


FIG. 9—TREAD RUBBER AVAILABLE FOR WEAR IN 36X5-IN. CUSHION TIRE OF 1700-LB. LOAD CAPACITY

This Type of Tire Occupies a Place in Deformability between the Solid and Pneumatic Tire. It Usually Has from 1½ to 2½ In. of Tread Rubber Available for Wear Varying with Sectional Width before It Will Split Open to the Internal Cavity and Will Give an Average Mileage of from 24,000 to 50,000 Miles, or from 4000 to 6000 Miles per ¼-In. of Wear

were determined by regularly inspecting and measuring the profiles of the tires on a large fleet of trucks and checking the wear against the mileage records of the individual tires. The table clearly and accurately reflects the markedly lesser rate of wear of cushion tires compared with solid tires.

Although the tire cost per mile with good cushion tires will not be very different from that obtained with good solid tires, the uninterrupted service that results from less frequent tire renewals, together with tractive characteristics that assure safety during the life of the tires, is much in favor of cushion tires. Fig. 10 shows in percentages the mileage that different types of tire must deliver to equal the tire-mile cost of single solid tires carrying equal loads. In all cases the performance of the single solid is considered as 100 per cent. Fig. 11 shows the condition of the cushion front-tires on a 5-ton truck of the Merchants Motor Freight fleet after running 52,000 miles between Philadelphia and New York City, Hartford, Conn., and northern Pennsylvania.

CUSHIONING PROPERTIES APPROACH THOSE OF PNEUMATIC TIRES

The necessity of cushioning fragile loads to the greatest possible extent is one of two reasons that per-

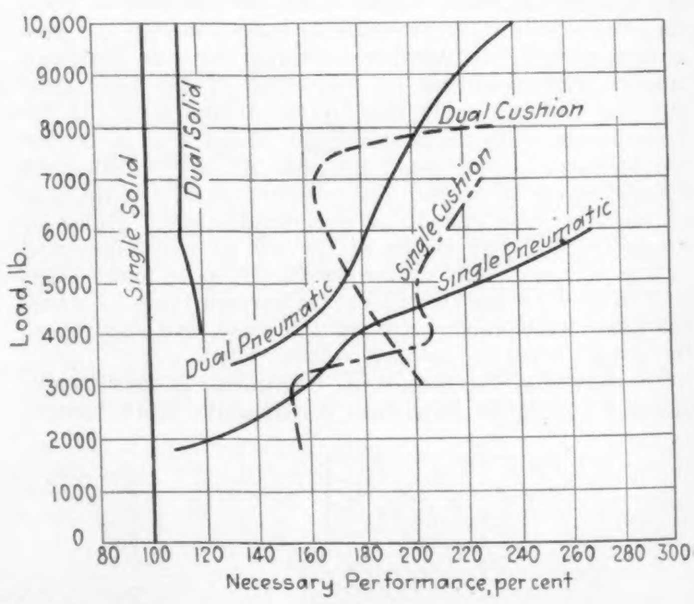


FIG. 10—MILEAGE PERFORMANCE OF OTHER TYPES OF TIRE REQUIRED TO EQUAL THE TIRE-MILE COST OF SINGLE SOLID-TIRES

The Per-Mile Cost of Solid Tires for Loads up to 10,000 Lb. Is Taken as 100 Per Cent. The Increased Mileage That Dual Solid, Single and Dual Pneumatic and Single and Dual Cushion-Tires Must Deliver To Be as Economical Is Shown in Percentages Reading to the Right at the Base of the Chart. Note the Decreasing Mileage Required of Dual Cushion-Tires for Loads of 3000 to 7000 Lb. and the Lower Mileage Required of Single Cushion-Tires for Loads from 5000 to 6000 Lb. Compared with the Single Pneumatic-Tires

haps justify the use of pneumatic tires on trucks. The other reason is the need of high speed. A good cushion-tire will provide sufficient cushioning to meet both requirements. A progress report[2] of general results on a series of road-impact tests that have been under way by a joint committee of the Society, the Rubber Association of America and the Bureau of Public Roads for more than 2 years was made at the Summer Meeting of the Society at French Lick Springs, Ind., in June, 1926. Detailed data have not yet been made public but this report shows clearly and definitely that a good cushion tire gives decidedly lower impacts than solid tires and that its cushioning qualities are nearer to those of the pneumatic

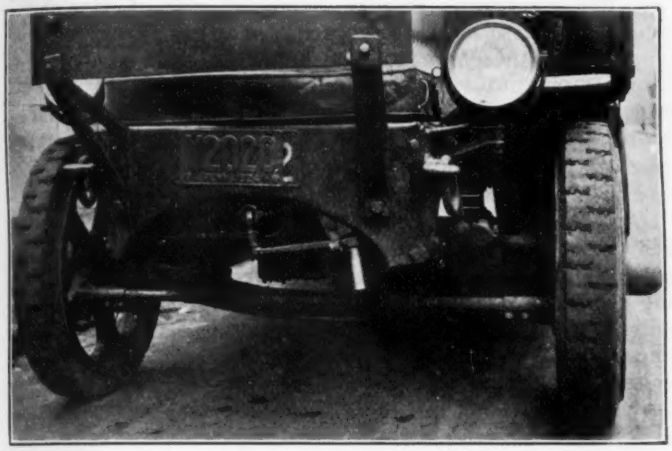


FIG. 11—CONDITION OF CUSHION FRONT-TIRES ON 5-TON TRUCK AFTER RUNNING 52,000 MILES

These Tires Ran 52,000 Miles on a 5-Ton Mack Truck of the Merchants Motor Freight Line between Dec. 2, 1922, and April 30, 1924, when the Photograph Was Taken. They Had then Made 215 Round Trips between Philadelphia and New York City, Two between Philadelphia and Hartford, Conn., and Several into Northern Pennsylvania

ment store operated 67 trucks on pneumatic tires during the year 1922 and the following year changed to cushion tires and operated 82 trucks on them. In that 1 year the saving on tires was $8,790, which included the wages of one tire man and one helper who were no longer required but takes no account of the avoidance of loss of time by the truck drivers and their helpers and delays in the delivery of goods that formerly occurred owing to the necessity of making pneumatic-tire changes on the road. When pneumatic tires were in use it was necessary to carry in stock 90 tires in three sizes, but with cushion tires the store has no investment whatever in spare tires. All pneumatic-tire casing repairs were made in the service stations of the several tire companies, which necessitated hauling the tires to and from the stations. The store is now operating 126 vehicles, mostly fast light-duty delivery cars, on cushion tires, but as no pneumatic tires have been used by it since 1922, no later comparisons of tire costs are available, but the garage superintendent is certain that the savings are very large.

When spare tires are carried on trucks they are not infrequently damaged, lost or stolen; such losses should be included in the records of tire-mile cost. The frequent use of chains necessary with pneumatic tires not only brings about heavy costs for the chains but is responsible for low mileage of many tires. The store found that with cushion tires the use of chains has been materially reduced with a corresponding reduction of cost and also that when used little if any damage has been done to the cushion tire.

REPAIR SAVINGS EFFECTED ON DUMP TRUCKS

The experience of a company that handles building materials and does contract hauling with a fleet of two hundred 5-ton dump trucks that carry average loads of 7 or 8 tons should be of interest as showing the savings

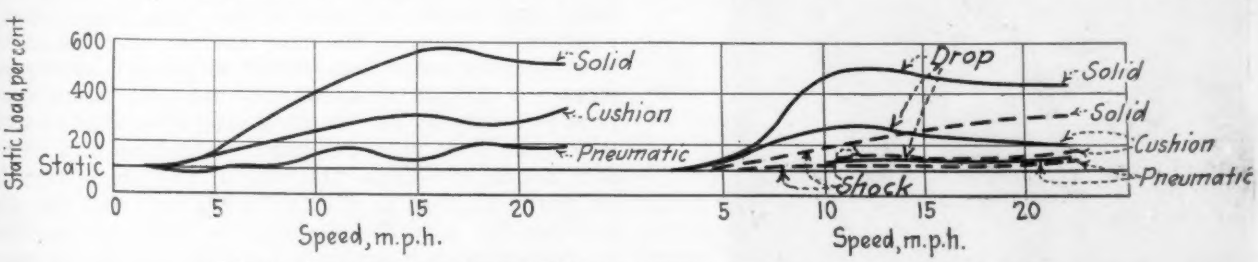


FIG. 12—RELATIVE CUSHIONING EFFECT OF SOLID, CUSHION AND PNEUMATIC TIRES

The Chart at the Left Shows the Shock in Per Cent of Static Load Due to a Drop of 1½ In. after Traversing a 30-In. Incline at Various Rates of Speed. That at the Right Shows the Shock at and Drop after Striking a Rectangular Block ⅞ In. High and 3 In. Wide. Data From Which the Charts Were Made Were Obtained in Impact Tests Conducted by the Bureau of Public Roads. The Tires Were Fitted on a 2-Ton Truck and Were Dual Tires of Rated Size Carrying Their Rated Load. In the Chart at the Right the Shock of Striking the Obstruction Is Indicated by Dotted Lines and the Impact of Drop by Solid Lines. Note the Close Approach of the Cushion Tires in Cushioning to the Pneumatic Tires at the Higher Speeds

tire than of the solid tire. So far as the results of these tests have been published, they show clearly that the cushion tire so effectively decreases impact that it must receive more favorable consideration by the operator who is interested in saving his trucks and by State highway commissioners who are interested in saving the roads. Fig. 12 is a reproduction of two charts presented in the report referred to and is presented here to show again the findings of the Bureau as to the relative position occupied by pneumatic, cushion and solid tires as regards road impact.

The light delivery wagon affords the best test of the effectiveness of the cushioning provided by the cushion tire as compared with that of pneumatic tires. Because of the light construction of the vehicle and the high speeds at which it is operated, the tire equipment is a big factor in the cost of its upkeep. A certain depart-

FIG. 13—TRACTION AFFORDED BY CUSHION TIRES WITHOUT CHAINS

FIG. 14—LOAD AND ROAD CONDITIONS UNDER WHICH THE TRACTIVE PROPERTIES OF CUSHION TIRES IS PROVED

that can be effected by the use of tires of the cushion type in classes of work that usually is done only on solid tires. The records of this company show that since using cushion tires during a 2-year period only two worm-drive rear-axle units have failed, whereas formerly it was necessary to overhaul a rear-axle unit every 6 months, due to road shocks from solid tires and the necessary jerking back and forth to help loosen loads that stick in the bodies. The approximate cost to the company of overhauling a rear-axle unit is $250.

While using solid tires the company always carried in stock from $200 to $300 worth of roller-bearings for the wheels, but it has not replaced a roller-bearing since it equipped its fleet with cushion tires and now carries only two spare bearings in stock for use in case of possible accident. The previous trouble with bearings cannot be attributed to improper maintenance, because its maintenance methods and greasing routine have not been changed.

The breakage of engine support-arms has also been reduced greatly. When one of these arms breaks an expense of from $150 to $200 or more is involved for repairs and the truck is out of service for 2 days at an expense of $35 per day.

FIG. 15—CUSHION TIRE DESIGNED FOR USE IN THE OIL FIELDS

This Is Called the "Mud Hen" and Gives Traction and Support to Heavy Loads in Roadless Areas and on Sandy Roads Where only Pneumatic Tires Could Be Used Formerly. The Contour Approaches That of the Pneumatic Tire when under Load and the Radial and Circumferential Webs Give Additional Ground Contact To Support the Load, While the Radial Webs also Give Additional Grip in the Mud. Although This Tire Violates All Accepted Principles of Standard Design, It Does not Chip Badly nor Undercut at the Base, as the Nose Is Very Flexible and the Rubber Deforms Easily When It Strikes a Hard Object

TRACTIVE PROPERTIES OF CUSHION TIRES

Cushion tires that are designed properly have excellent tractive and non-skid qualities. In localities where weather conditions require extensive use of tire-chain equipment on trucks fitted with solid tires, the cost of the chains amounts to a considerable figure in a year and, more important still, the solid tires are so badly cut by the chains by the end of the winter that their replacement becomes necessary long before it should be. Examples of the traction afforded by cushion tires without the use of chains are given in Fig. 13 and 14.

Cushion tires are well adapted to molding in special designs for special purposes. We have experimented for a long time to produce a form of tire suitable for carrying a truck through the roadless oil-fields where formerly only pneumatic tires could give the necessary performance. Solid and cushion tires of standard design acted as rut diggers or wedges and caused the rear wheels to become mired. A cushion tire of special design, shown in Fig. 15 and dubbed the Mud Hen, approximates in contour the shape of a pneumatic tire when under load. The radial and circumferential webs give additional ground-contact to support the load, and the radial webs also give additional tractive grip in mud. This tire violates every principle of accepted standard design; according to previously accepted criteria it should chip badly and should undercut at the base, but it does not seem to do either. We have records of some of these tires running more than 20,000 miles and, although the webs are slightly abraded, they are still intact and the tires are carrying the trucks through the kind of soil for which the tires were designed. Abrading and chipping does not occur because the nose of the tire is very flexible due to its high profile and to the fact that the design provides for easy displacement or local deformation of the rubber when it strikes a hard object, thereby spreading the load over a large portion of the base. Quite by chance we found that the tire gives very satisfactory service on sandy roads, which constitute another type of road that commonly is traversed only with giant pneumatic tires on trucks.

In some cases the users of cushion tires fit tire chains in bad weather merely as assurance against a possible charge of negligence in event of a claim for damages. However, when chains are used with this type of tire the wheel is not forced up off the road surface each time the tire rolls over the cross links of the chain because the softness of the tire allows it to absorb within the tread the links of the chain and minor road irregularities. This characteristic gives much longer chain life and also avoids excessive cutting of the tire by the chain.

INDIRECT SAVINGS EFFECTED BY CUSHION TIRES

Avoidance of accidents is of increasing importance. Shop repairs and insurance premiums are reduced correspondingly as accidents are reduced. Because of their greater flexibility and more effective tread design,

(*Concluded on p. 155*)

Discussion of Papers at the 1926 Semi-Annual Meeting

THE discussion following the presentation of two of the papers at the Semi-Annual Meeting of the Society that was held at French Lick Springs, Ind., in June, 1926, is printed herewith. The authors were afforded an opportunity to submit written replies to points made in the discussion of their papers and the various discussers were provided with an edited transcript of their remarks for approval before publication.

For the convenience of the members a brief abstract of each paper precedes the discussion so that members who desire to gather some knowledge of the subjects covered without referring to the complete text as originally printed in the June 1926 issue of THE JOURNAL, can do so with the minimum effort.

INSTRUMENTATION AND RESULTS OF RIDING-QUALITIES TESTS

BY ROY W. BROWN[1]

ABSTRACT

ULTIMATE successful measurement of riding-qualities is indicated by the design and construction of several new accelerometers. Experimental work with these new instruments has served to emphasize the desirable and the undesirable instrument-characteristics essential to accuracy, and thrown light upon some conflicting results heretofore obtained. One of the chief detriments to the development of accurate riding-qualities instruments has been the lack of satisfactory calibration. In the past, space-time curves have been used largely for calibration but, from the viewpoint of accuracy, these curves generally are acknowledged to be unsatisfactory.

An elaborate machine has been constructed which will produce simple harmonic-motion to a very high degree of accuracy. This machine provides a convenient and highly accurate method of calibrating accelerometers in terms of known fundamentals and, in combination with a method for inter-comparing different instruments under simulated road-conditions, it enables instrument calibration through the entire range of vibrations caused by road irregularities.

Measurements of axle accelerations and displacements indicate the need of concentration at this time on the correlation of the various factors entering into riding-qualities. However, a finite evaluation cannot be reached without investigation of physical and mental effects. The need for such an investigation is apparent.

THE DISCUSSION

B. LIEBOWITZ[2]:—Contact accelerometers are destined, I believe, to play an important role in the riding-qualities problem, and some knowledge of the history of this type of accelerometer is of interest.

The earliest reference I found is a paper entitled Le Accelerometre A Maxima Du Laboratoire D'Essais Du Conservatoire National Des Arts E Metiers[3]. A full-fledged design of what we would now call a "single-element adjustable contact-accelerometer" is shown in that paper. In view of the simplicity and directness of the method, it seems possible that earlier references may be found; if not, then Auclair and Boyer-Guillon deserve the credit for the invention of the contact accelerometer. Their instrument was described in several French publications[4]; also, in at least one English publication, the reference to which I am unable to recall.

The credit of first describing, in 1919, a *battery* of these cells arranged so as to draw a discontinuous curve of the rise and fall of acceleration appears to belong to Zahm.

In the contact type of accelerometer, it is the *initial motion* that is employed for measurement. Until very recently, efforts in this Country to develop instruments for the direct measurement of acceleration were confined to other types, mainly to spring-weight combinations the *maximum motions* of which were intended to represent the acceleration, as is well brought out in a paper by J. A. C. Warner entitled Riding-Qualities Research[5]. In the discussion of Mr. Warner's paper[6], I pointed out the difficulties with the usual spring-weight combination and called attention to the advantages of the contact principle.

Mr. Brown's paper shows four designs of contact accelerometer, and it is of interest to point out their relative characteristics. Brown's first instrument[7], shown at the top in Fig. 1 of his paper, is a pioneer design and has served a highly important purpose as such. From a practical viewpoint, its disadvantages lie in its excessively stiff spring and expensive construction.

Dr. H. C. Dickinson's diaphragm-type design[8], shown at the bottom in Fig. 1 of Mr. Brown's paper, is fundamentally very reliable, has an open scale and is practically free from error due to local vibration of any of its parts. Its weight and bulk, which are an asset in certain intended applications, do not lend it to battery construction or to uses in which lightness and compactness are required.

In designing the micrometer type[9], shown at the center of Fig. 1 in Mr. Brown's paper, I aimed at simplicity and compactness, with adaptation to battery construction in view. The only part susceptible to local vibration is the

[1] In charge of engineering laboratories, Firestone Tire & Rubber Co., Akron, Ohio.

[2] M.S.A.E.—Reading, Pa.

[3] See *Memoires de la Societe Ingenieurs Civils de France*, by Auclair and Boyer-Guillon, 1913, vol. 2, p. 51.

[4] See *Comptes Rendus des Seances de l' Academie des Sciences*, vol. 169, p. 24; see also *Bulletin de la Societe d' Encouragement pour l' Industrie Nationale*, vol. 134, p. 177.

[5] See THE JOURNAL, July, 1924, p. 75.

[6] See THE JOURNAL, December, 1924, p. 558.

[7] See THE JOURNAL, December, 1925, p. 546.

[8] See THE JOURNAL, March, 1926, p. 249.

[9] See THE JOURNAL, March, 1926, p. 250.

flat spring itself, but this is designed as a reed having a fundamental natural period shorter than any likely to be encountered in service; hence, little error is to be expected from that source. This type of construction is relatively inexpensive.

In his latest designs, shown in Figs. 2 and 3 of his paper, Mr. Brown has produced a Zahm type of highly refined construction, light and compact and having exceptionally good electric-contact mechanism. While it has more parts susceptible to local vibration in the instrument itself, this should not interfere in its intended field of application. Mr. Brown's 10-element design is a noteworthy example of battery construction.

I wish to emphasize the wide field of application of such accelerometers as Mr. Brown has mentioned. First, it should be noted that they can be used to measure *longitudinal* as well as vertical road-shocks and also lateral road-shocks. The calibration changes by $1g$ when the instrument is turned on end. Then, the single-element type would be very useful in exploring engine vibrations, on the engine, on the chassis and in the body. Comparative measurements of "knock" might also be made by this means, and other uses, both inside and outside the automotive field, might be enumerated. The contact-accelerometer principle places an excellent tool of investigation at the engineer's disposal.

CHAIRMAN J. W. WHITE[10]:—Will this instrument become available to manufacturers for experimental work in developing cars or for developing other things such as rebound-control devices, cushion springs and the like?

R. W. BROWN:—It is our purpose to make the results of our efforts available to whoever desires to take them up.

J. H. HUNT[11]:—Should not the displacement curve be 180 deg. ahead of the acceleration curve?

MR. BROWN:—That is correct.

F. E. MOSKOVICS[12]:—It seems to me that there is a chance for interconnection between the control of the rate of the vehicle body and the rate at which the passenger is allowed to rise with the air-vent in the seat open.

QUESTION:—In reference to the chassis set-up and the drum, does the wheel drive the drum?

MR. BROWN:—Yes, but merely as a matter of convenience. I understand that in some similar set-ups the drum drives the wheel. We arrange the set-up so that we can remove one chassis and put another chassis over the drum with the minimum amount of work, and that involves driving the drum. In fact, the set-up is arranged so that we can mount any commercial truck or passenger-car directly over the wheel, attach our instruments and be ready to take readings on that particular car.

Engineers who have had experience with drum dynamometers in fleets may say that it requires a relatively large drum to permit sufficient time to allow the vibrations to die out before the obstruction passes around the drum and again comes under the tire. In this case the drum is an exceedingly heavy cast-steel flywheel which is approximately 6.6 ft. in diameter and has a circumference of about 20.7 ft. In all ordinary cases, up to speeds of 20 or 25 m.p.h., with solid and with cushion tires, and up to speeds of 30 or 35 m.p.h. with pneumatic tires, the vibrations are virtually damped out beforehand.

W. D. SEED[13]:—For those who are more interested in the chassis spring-suspension where a less rugged type of accelerometer can be used, the International Motor Co. uses an accelerometer which gives a graphic record. The spring for this accelerometer has a natural period over five times the period of our motorcoach springs. An optical attachment equipped with a roll of moving-picture film gives a record of acceleration. At the same time these accelerations are recorded, we use a position recorder to measure the displacement of the axle with relation to the body of the motorcoach. These measurements are taken primarily over standard bumps at certain set speeds. This gives a direct comparison when springs are changed and also gives a good physical record of the experiments that have been performed without relying primarily on the impression of the observer. In addition to these two instruments, an integrating accelerometer is used for finding the amount of acceleration over a measured mile of various types of road at certain set speeds. This has been found to give the best relative measure of riding quality. Comparative readings between motorcoaches and the best types of passenger-car can be obtained by using this method.

BALLOON TIRES FOR USE WITH DROP-CENTER RIMS

BY B. J. LEMON[14]

ABSTRACT

BALLOON tires for drop-center rims will further complicate the present muddled balloon-tire wheel-and-rim situation. The complications for the tire industry will be less than those for the wheel and the rim industries. Drop-center tires and rims with detachable wheels are the vogue in Great Britain; the movement is spreading to Continental Europe and is felt in America. British practice in tire construction has abandoned the "catch" bead; in rim dimensions, it has adopted the flange height and width of American flat-base rims. This is a step toward international rim-standardization. British tire diameters are trending toward 19 and 21-in. sizes. An apparent tendency exists to reduce the well-depth in drop-center rims. Medium-pressure tires are obtainable by such British builders of heavy automobiles as are not satisfied with the larger and more flexible balloon-tires.

The drop-center rim first came into use in America for the airplane, on which this rim is now standard. The Ford steel-spoke wheel with 4.40-in. drop-center tire and rim is the only American development to reach the commercial stage. A combination 4.40-in. casing and tube are obtainable commercially that will fit both 3½-in. flat-base and 4.40-in. drop-center rims. By convexing the well-slopes of the drop-center rim inward, more satisfactory conditions for proper seating of tire beads on rim bead-ledges are obtained. To render visible this bead seating a red guide-line is placed above the rim flange on the outside of the 4.40-in. drop-center casing. Such advantages and disadvantages as

[10] M.S.A.E.—Chief engineer, Wire Wheel Corporation of America, Buffalo.
[11] M.S.A.E.—Head of electrical division, General Motors Corporation Research Laboratories, Detroit.
[12] M.S.A.E.—President and general manager, Stutz Motor Car Co. of America, Inc., Indianapolis.
[13] International Motor Co., New York City.
[14] M.S.A.E.—Automotive contact engineer, United States Rubber Co., Detroit.

are now apparent are listed for 4.40-in. drop-center tires and rims. Instructions are given for the applying and dismounting of 4.40-in. tires on drop-center rims.

Development in America of wheels, rims and tires in sizes larger than 4.40 in. on the drop-center principle has been half-hearted. Results thus far with larger tires on drop-center rims are inconclusive. Two American wheels with drop-center rims are described. The apparent need for a straight-side tire and rim for motorcycles, perhaps of drop-center type, is imperative. The question of preference by the American motorist and car builder for a demountable-type rim, the degree of success of the Ford drop-center venture and the influence of export demands are cited as bearing heavily on the permanency of success of the drop-center tire and rim in America.

The Discussion

H. W. KRANZ[15]:—Prior to the World War, the drop-center-type rim, which was originally developed in Europe, was not used in any instance within our knowledge in America. During the World War a few were manufactured and used in Europe, as we understand it, for experimental and development purposes only, but we do not know how many. Neither do we know whether some of these rims were used in an experimental way on airplanes. Starting early in 1917, and after the War until the end of the year 1919, the type of rim used for airplanes in America was what is known as the Palmer-type rim. A cross-section of this rim, which was adopted by the S.A.E. Standards Committee on Dec. 14, 1917, is shown in Fig. 1, together with data pertaining to it. Fig. 2 shows the way the clincher tire seats when inflated on these rims.

[15] Cleveland Welding & Mfg. Co., Cleveland.

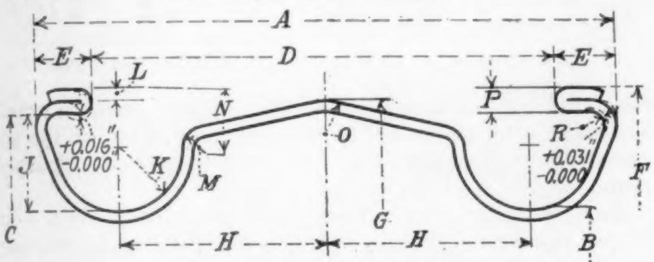


FIG. 1—CROSS-SECTION OF THE RIM ADOPTED BY THE S.A.E. STANDARDS COMMITTEE

The Rim Shown Was Adopted on Dec. 14, 1917. The Cold-Rolled-Steel Specification Is

	Per Cent
Carbon	0.18 to 0.25
Manganese	0.35 to 0.55
Phosphorus	Less than 0.05
Sulphur	Less than 0.05

Other Data Relating to This Rim Are Tabulated Below

Rim Mark		Z_2		Z_3	
Thickness, In.	S.W.G.	17.5		15.0	
		In.	MM.	In.	MM.
	....	0.052	1.321	0.072	1.829
Size		27.56x3.94 29.53 4.93	700x100 750x125	31.5x5.91	800x150 ....
Diameter	B	18.826	479.6	18.826	479.6
	C	20 039 +0.032 −0.000	508.6	20.039 +0.032 −0.000	508.6
	F	20.315	515.6	20.315	515.6
	G	20.173	512.0	20.173	512.0
Circumference	F	63.828±0.046[a]	1620	63.828±0.046[a]	1620
	G	63.355	1608	63.355	1608
A		3.386	86	3.386	86
D		2.677±0.046[a]	68	2.677±0.046[a]	68
E		0.354	9	0.354	9
H		1.181	30	1.181	30
J		0.571	14.50	0.571	14.50
K		0.374	9.50	0.374	9.50
L		0.069	1.75	0.069	1.75
M		0.118	3.00	0.118	3.00
N		0.335	8.50	0.335	8.50
O		0.492	12.50	0.492	12.50
P		Max. 0.104 Min. 0.093	Max. 2.64 Min. 2.36	Max. 0.144 Min. 0.133	Max. 3.66 Min. 3.38
R		0.079	2	0.079	2

[a] Tolerances subject to Tire and Rim Association.

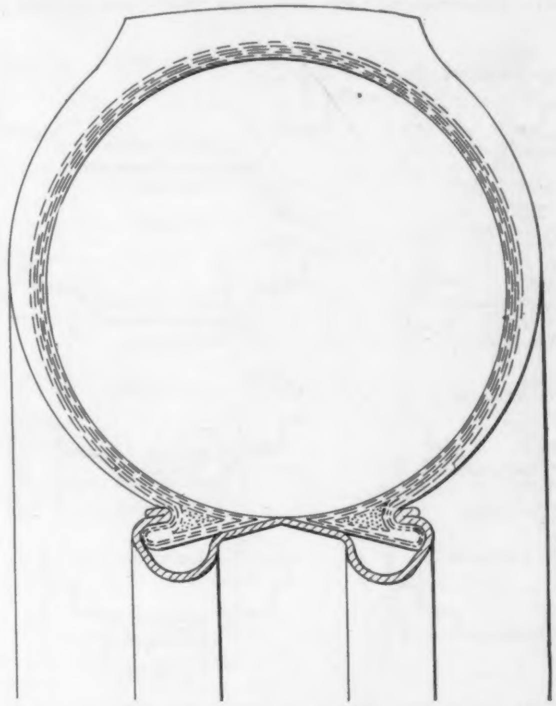


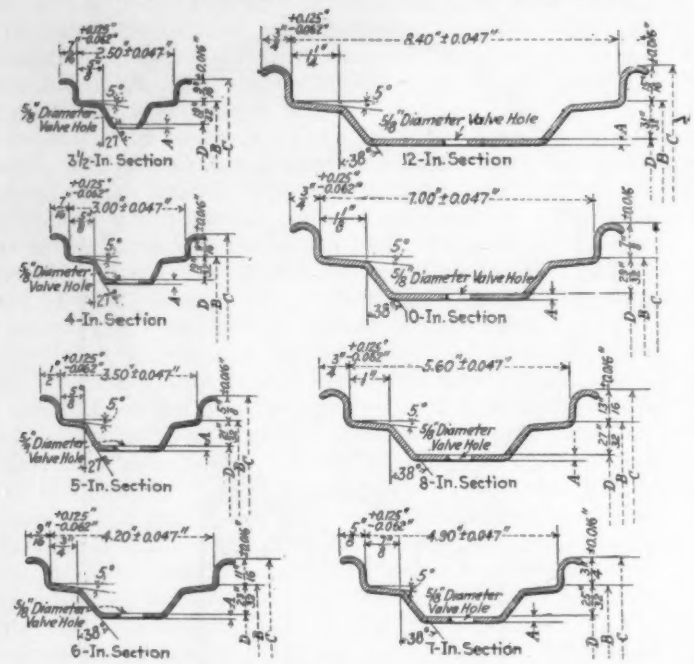
FIG. 2—SEATING OF A CLINCHER TIRE

The Manner in Which the Clincher Tire Seated on the Rim Shown in Fig. 1 Is Illustrated

Early in 1920, W. H. Allen, of the B. F. Goodrich Co., showed us a sketch of drop-center rims, later identified as the American type of drop-center rim, as illustrated in Fig. 3. By referring to this illustration it can be seen readily that the construction of this rim design differs in contour insofar as the contour is constructed in a similar manner to that of the present rim-standard adopted in America. Mr. Allen's intentions were to have rims manufactured to this design and, with the aid of his company, which would furnish the tires, to endeavor to interest the Government's aviation officers at McCook Field, Dayton, Ohio.

In May, 1920, we completed a few 36 x 8-in. rims for the Goodrich Co., which were later made up into wheels and shipped to McCook Field. The aviation officers were very much pleased with the new design as it had the advantage of lighter weight and enabled the use of straight-side tires; in fact, the enthusiasm grew to such an extent that on June 14, 1920, the Tire and Rim Association brought this matter before its meeting for discussion, the result being a decision that two other sizes should be manufactured. After a period of development, the matter was again discussed and this resulted in an adoption by the Tire and Rim Association on May 18, 1921, of this type of rim for American airplanes. This rim is still being manufactured for airplane service and,

DATA RELATING TO THE AMERICAN TYPE DROP-CENTER RIM



The Design Differs in Contour Compared with That of the Present Rim-Standard Adopted in America. Data on the Different Sizes Are Given in the Table in the Next Column

in fact, is the only construction being used in America at present in this service.

Manufacturing American-Type Drop-Center Rims

The method used by our company in manufacturing the airplane drop-center rim, which is also identified as the American-type drop-center rim, is as follows: Flat hot-rolled strip-steel or special hot-rolled sections are used as raw material and this material is formed into hooped or circular shape; it is then welded and the burr of the weld is removed. After being made into this form, the hoop is put through special rim-rolling machines equipped with guides to assure proper guiding of the material and caused to be cold-formed or rolled through a series of operations which are shown in Figs. 4 and 5.

The manufacturing methods for the American-type drop-center rim, as described, prove to be fairly simple and it is still being manufactured in this manner when-

Size, In.	Thickness, In.	Weight, Per Rim, Lb.	Diameter, In.	Circumference, In.	
	A		B	C Tolerance ±0.047 In.	D Tolerance ±0.062 In.
27x3½	0.083	6.52	20	62 53/64	59 3/32
	0.109	8.56	20	62 53/64	59 3/32
28x4	0.083	7.56	20	62 53/64	59 3/32
	0.109	9.93	20	62 53/64	59 3/32
30x5	0.094	10.02	20	62 53/64	58 45/64
	0.125	13.33	20	62 53/64	58 45/64
32x6	0.109	13.29	20	62 53/64	58 5/16
	0.125	15.28	20	62 53/64	58 5/16
34x7	0.115	16.13	20	62 53/64	57 59/64
	0.140	19.60	20	62 53/64	57 59/64
36x8	0.125	19.72	20	62 53/64	57 17/32
	0.156	24.61	20	62 53/64	57 17/32
44x10	0.141	31.39	24	75 13/32	69 45/64
	0.175	38.96	24	75 13/32	69 45/64
48x12	0.156	40.22	24	75 13/32	69 5/16
	0.175	44.80	24	75 13/32	69 5/16
54x12	0.156	50.25	30	94 1/4	88 5/32
	0.175	56.55	30	94 1/4	88 5/32

Data Relating to the American Type Drop-Center Rim

ever this type of rim is called for. Fig. 6 shows a cross-sectional view of an airplane drop-center rim or the so-called American-type drop-center rim cut from a finished rim.

Early in 1921, our company manufactured a 32 x 4-in. drop-center rim similar in design to the rim adopted by the Tire and Rim Association for use in connection with high-pressure tires on automobiles. Sample wheels were made up, but the B. F. Goodrich Co. did not make tires for these wheels and, as it was impossible to use the standard tire due to the limited width of rim and the considerably larger width of tire bead, both the Goodrich Company and our company discontinued developments of this type of rim in connection with high-pressure tires.

The Dunlop Rubber Co., Ltd., of Birmingham, England, the originator of this type of rim, had experienced trouble in the course of development from having tires leave the rims when in a deflated condition. To overcome this difficulty, the company had developed what it called a "catch bead," which is an extension located over the outer curve of the rim contour. This, the company claimed, overcame this difficulty. In the summer of 1923,

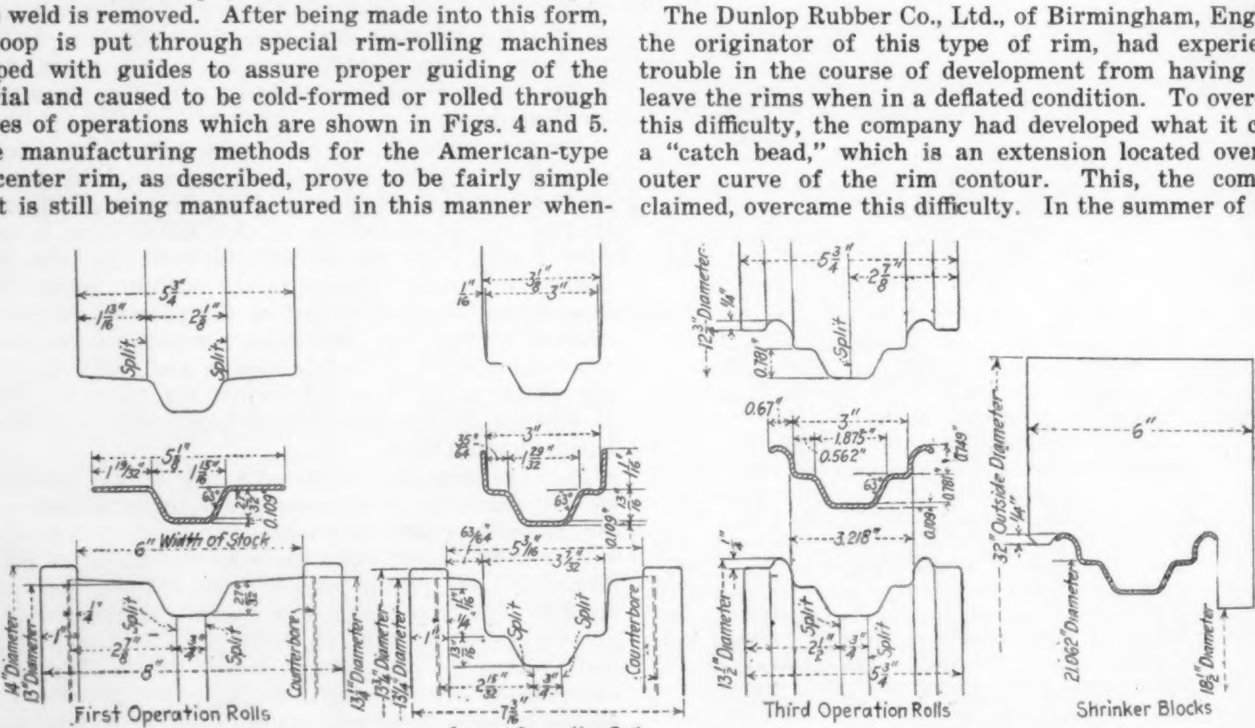


Fig. 4—Manufacture of American-Type Drop-Center Rims
The Raw Material Is Formed or Hooped into Circular Shape; It Is Then Welded and the Burr of the Weld Is Removed. The Hoop Is Then Put through Special Rim-Rolling Machines Equipped with Guides To Assure Proper Guiding of the Material and Is Caused To Be Cold-Formed or Rolled through the Series of Operations Indicated in This Illustration and Those Shown in Fig. 5

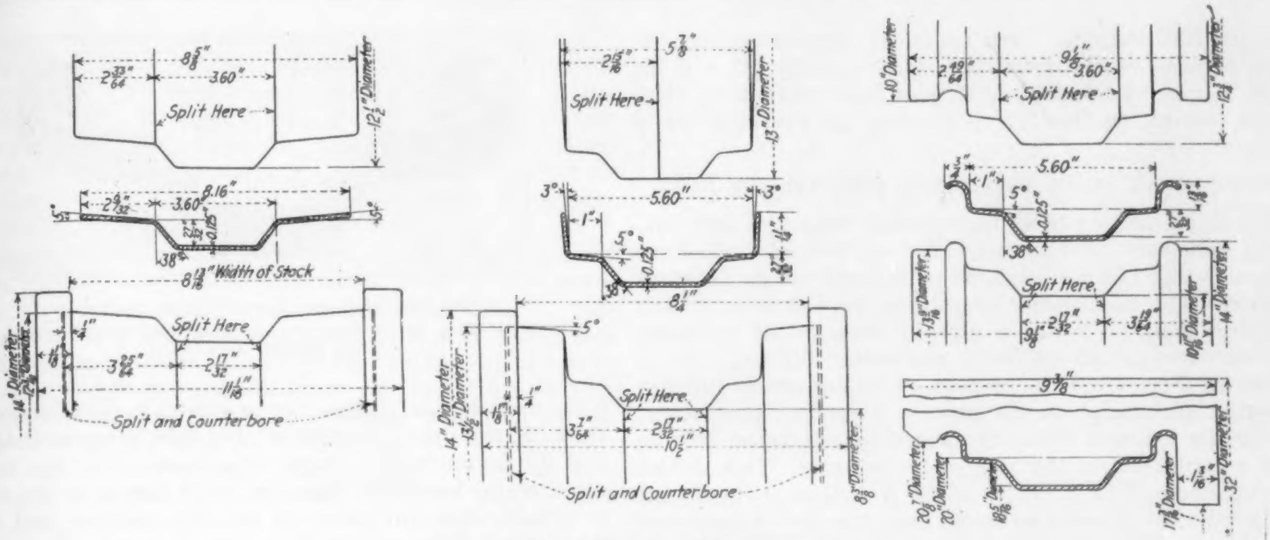


FIG. 5—DROP-CENTER-RIM MANUFACTURE
The Operations Indicated in Fig. 4 Are Continued in This Illustration

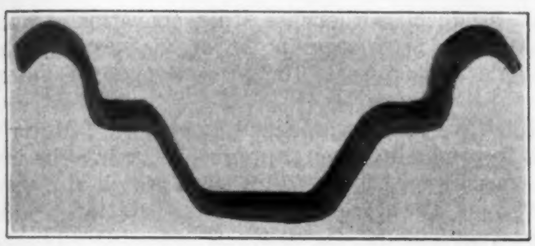


FIG. 6—CROSS-SECTION OF DROP-CENTER RIM
This Is a View of an Airplane Drop-Center Rim, or the So-Called American-Type Drop-Center Rim, cut from a Finished Rim

Colin Macbeth, of the same Dunlop Company, came to America to encourage and to instill enthusiasm into American tire makers for the drop-center rims, and he described the various merits the drop-center rims had over those of the present rims in general use. The entire trade is familiar with Mr. Macbeth's visits. In the latter part of 1923, three sizes of drop-center rims were produced by our company for the Budd Wheel

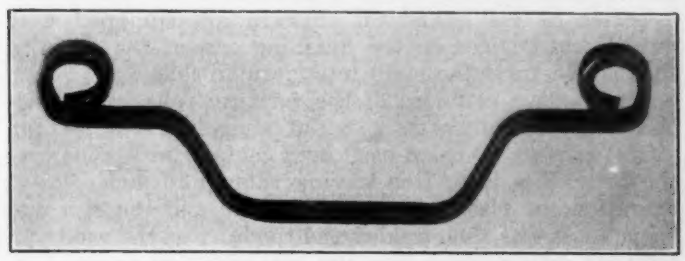


FIG. 8—DEFECT OF THE PROCESS SHOWN IN FIG. 7
The Tire Flanges Are Curled Inwardly in Both Cases, But It Was Not the Intention for the Rim To Be So Manufactured

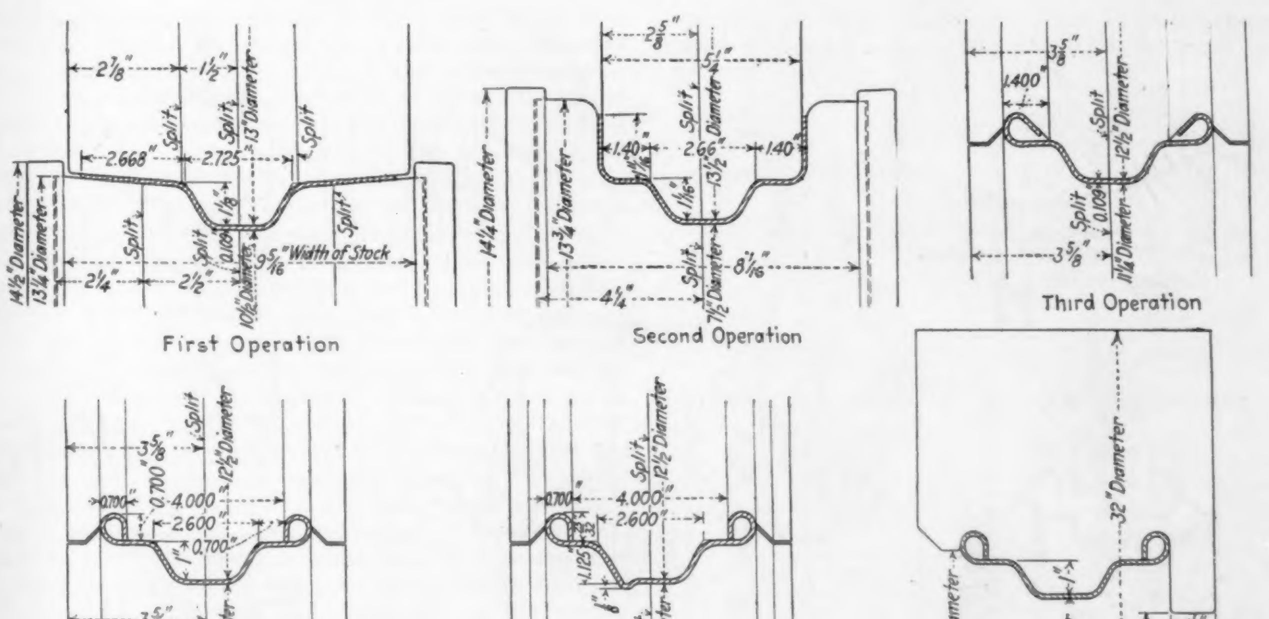


FIG. 7—MANUFACTURE OF INVERTED-BEAD DROP-CENTER RIMS
The Cold-Rolling Operations Were Done in Rolling-Machines Similar to Those Mentioned for Figs. 4 and 5, But the Various Operations Are Somewhat Different

Co., of Philadelphia. We produced approximately 150 rims of each of the following sizes; namely, 32 x 6 in., 34 x 7 in., and 36 x 8 in. These rims were very similar in design to the European type of inverted rolled contour.

Manufacturing Inverted-Bead Drop-Center Rims

For the inverted-bead drop-center rims the raw material consisted of flat, hot-rolled strip-steel formed or hooped, welded and burred, as mentioned in the previous method. The cold-rolling operations were done in rolling machines similar to those already mentioned; however, the various operations were somewhat different as is shown in Fig. 7. This process of manufacture did not prove as successful as the process described previously, due to the extreme difficulty caused by not being able to hold or conform to the required contours. This defect is explained by Fig. 8, which shows a section of the actual rim produced. It will be noted that the tire flanges are curled inwardly in both cases, but it was not the intention for the rim to be so manufactured. However, this method of manufacture did not overcome the curling; therefore, it had to be abandoned to make way for a different process.

In this subsequent process, the raw material was a flat open-hearth-steel strip fed into what is commonly known as a multiple forming-machine which consists of a series of roll standards, five in all, equipped with mechanical devices on the finishing end of the machine that would form the strip into circular shape.

The various contours of the forming rolls in the machine which transforms this flat strip-steel bar into an almost perfect inverted-bead drop-center-rim section, are shown in Fig. 9. After leaving this rapid and efficient operation the rim contour was formed and the rim was hooped. It was then welded and the burr of the weld was removed, after which the rim was sized to proper circumferential measurements.

The foregoing method produced a far better inverted-

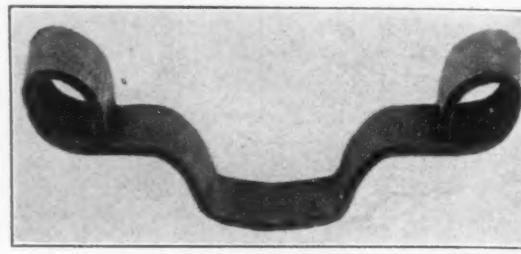


Fig. 10—Section from a Finished Rim
This Rim Is at Present Manufactured in Comparatively Large Quantities

bead drop-center rim so far as contours were concerned than did the previous method; but this latter method also had its faults in that these rims were noisy due to oxidized metal breaking from the weld burrs on the inside or tubular-shaped portion of the rim contours and, after these rims were mounted in wheels, even when rolled across a floor, they would give a zither-like noise which had to be overcome before they could be accepted as a perfect product. Otherwise, this operation proved efficient from a productive point of view and equally as reasonable in cost as is the present type of automobile rim.

In 1924, we were approached by the Ford Motor Co. in regard to producing drop-center rims having the inverted rolled contour for 29 x 4.40-in. tires, which rim was later approved by the Tire and Rim Association. This rim is at present manufactured in comparatively large quantities and is being used on wire-wheel equipment for Ford cars. Fig. 10 shows a section cut from a finished rim manufactured under the process just described.

The next process of manufacture developed and in use today is as follows: The flat open-hearth-steel strip is formed or hooped, is welded and has the burr removed as described in processes Nos. 1 and 2. This plain band is then put through a press operation, during which the

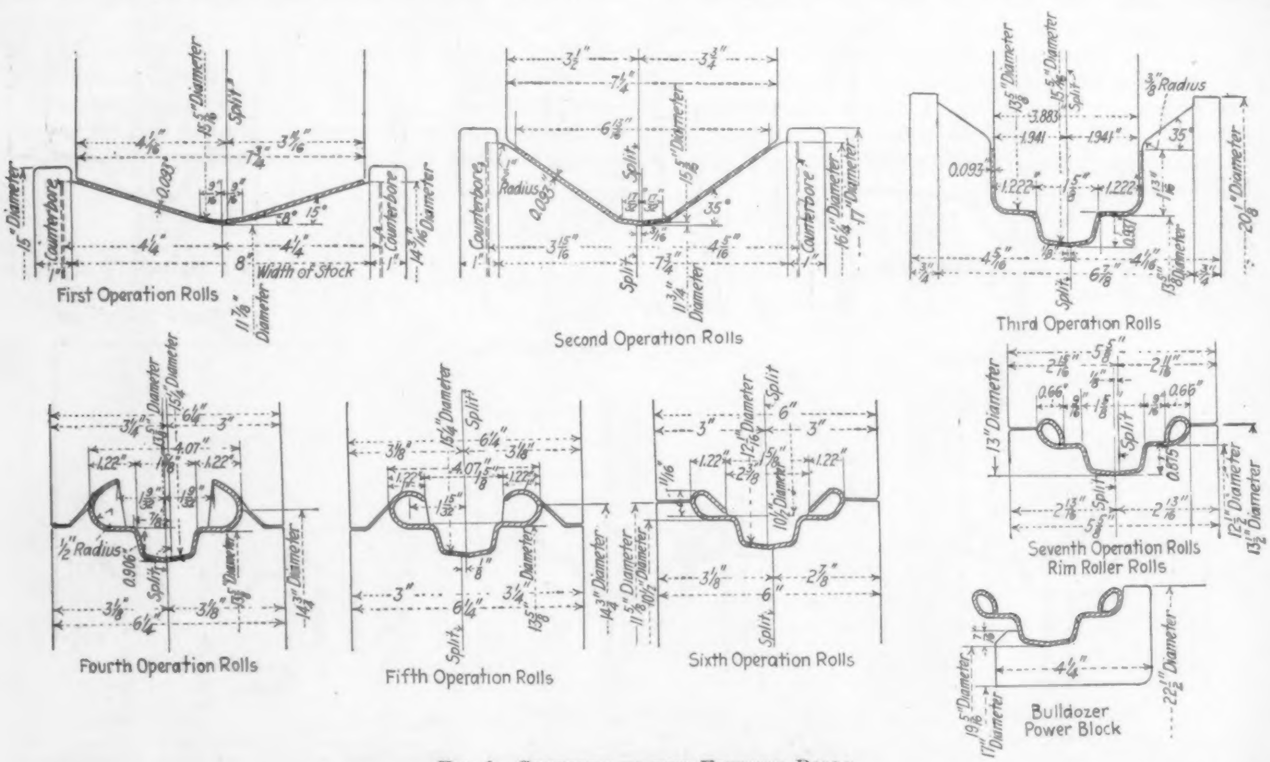


Fig. 9—Contours of the Forming Rolls
The Various Contours of the Forming-Rolls Transform the Flat Strip-Steel Bar into an Almost Perfect Inverted-Bead Drop-Center-Rim Section

two edges of this band are curled. The curled edges are formed practically to the required contours of the rim when finished. This operation is shown in Fig. 11. After the completion of this operation, the material is run through a rim-rolling machine similar to that described previously, in which operation the drop well is rolled as shown in Fig. 12, after which it is sized, making the completed rim as shown in Fig. 10.

The method just outlined proves to be very efficient and a considerable number of inverted-bead, or European-type, drop-center rims have been manufactured in this manner. This eliminates all troubles and difficulties heretofore encountered and enables the rim to be manufactured within the specifications of the Tire and Rim Association of America. The method produces an entirely satisfactory rim, as the metal is not strained due to mechanical operations. This process enables us substantially to maintain present-day rim-costs on inverted-bead drop-center rims.

Drop-Center Tire-Equipment

J. W. White[16]:—Although the drop-center rim and tire were introduced into this Country more than 2 years ago, there has been no general adoption of it. Only one manufacturer adopted it, and that was in a more or less experimental way. As a result, a question exists in the mind of the automotive engineer as to why this program has not received more favor in the United States; however, upon investigation of the conditions here as compared with those in England, the reason is a natural one. In England, this development was sponsored by a company which makes tires, rims and wheels. One of its largest competitors also makes tires, rims and wheels, thereby bringing about competition in the construction and sale of the wheel as a unit with the tire. The condition in this Country is just the reverse. Here, tire making is a separate industry and even rim making and wheel making are more or less separate. There is no inducement for the tire companies to sponsor the development of drop-center tires because it means a change in molding equipment, particularly on the part of those who use the flat braided-wire bead which inherently requires a wider base-section than that of the cable-type-bead tire. The cable-type bead is used by three of the major tire manufacturers, and the three others have generally used the flat braided-wire bead. To the tire manufacturers, it yields nothing in tire performance or sales over and above those of the present balloon-tire, with the bare possibility that the company which advocates it may suffer in pioneering it and, if it were successful, their competitors would be equally free and quick to adopt it.

The rim manufacturer is not vitally interested, because it does not mean any more rim business but does involve new methods of rolling and, consequently, new rolling equipment. Inasmuch as material is the major item in the cost of a rim and labor is the smallest item, the rim manufacturer most naturally will prefer to make the rim which his steel sources are equipped to roll. The wheel manufacturer is influenced by the same reasons and, in addition, if he pioneers the development, his capital outlay will be enormous because it will involve different disc or spoke requirements for the same wheelbase in addition to his present equipment and, inasmuch as his profits depend upon long runs on one size of wheel, he is loath to advocate that which does not promise large production in the immediate future. From the foregoing it will be seen that, if the American tire-manufacturers were also

[16] M.S.A.E.—Chief engineer, Wire Wheel Corporation of America, Buffalo.

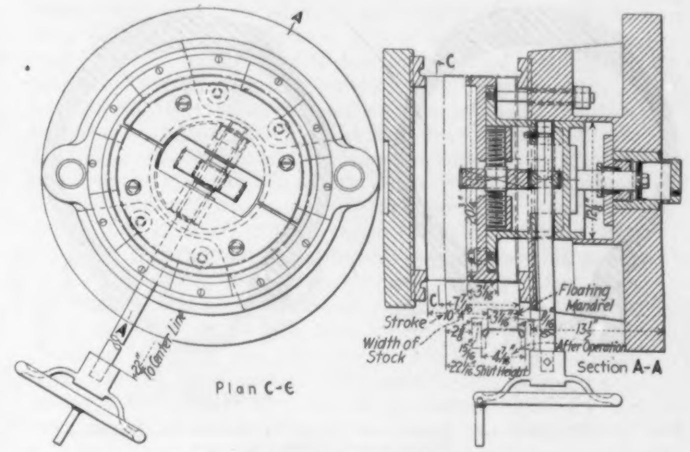


Fig. 11—Rim-Curling Fixture
The Flat Open-Hearth-Steel Strip Is Formed or Hooped, Is Welded and Has the Burr Removed. This Plain Band Is Then Put through a Press Operation, during Which the Two Edges of This Band Are Curled. The Curled Edges Are Formed Practically to the Required Contours of the Rim when Finished

making rims and wheels as in Europe, thus competing with other manufacturers similarly equipped, there would be a real incentive. The real advantage in the adoption of a drop-center design, if successful, accrues mainly to the car builder.

With the advent of four-wheel brakes and balloon tires, the necessity for keeping unsprung-weight down has become much more important due to the increased tendency of the car to shimmy as a result of increased tire-flexibility, lighter springs and the increased unsprung-weight involved by the use of a brake on the front axle. I believe that, when properly developed, the drop-center equipment is inherently lighter in weight than the present flat-base-rim type of construction and, because it is a one-piece rim, it offers much in the line of simplicity, ruggedness and ease of assembly over the present type, with a corresponding reduction in cost.

Defects of Drop-Center Tires

In view of their possible adoption in this Country, it is well to analyze some of the defects which have become

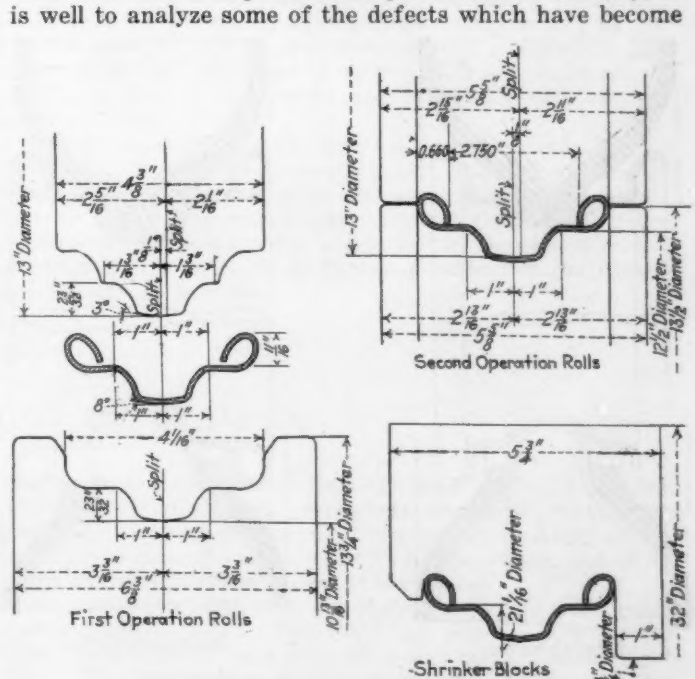


Fig. 12—Rolling of the Drop Well
After the Drop Well Is Rolled, the Rim Is Sized. It Is Then in the Completed Form Shown in Fig. 10

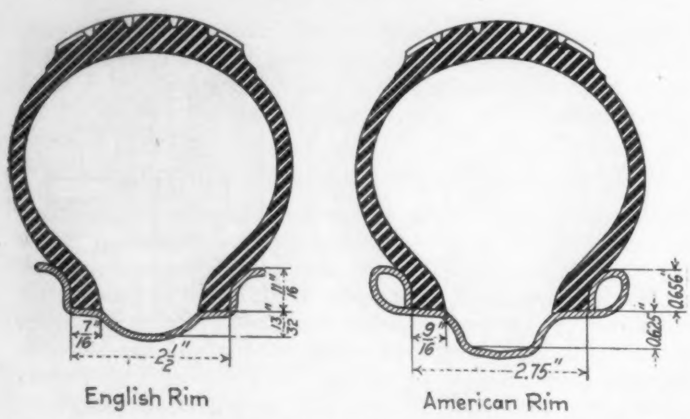


FIG. 13—ENGLISH AND AMERICAN RIMS WITH 4.40-IN. BALLOON-TIRE EQUIPMENT COMPARED

The Well on the English Rim Is Much More Shallow Than That on the American Rim; Consequently, a Smaller Tube Can Be Used with the English Rim and Less Difficulty Is Experienced in Getting It To Fit the Well

evident in drop-center tires and rims as used experimentally in this Country thus far. The first difficulty, and the most important one, is tube pinching. The drop-center rim requires a tube of larger cross-section and smaller diameter than is used in the flat-base construction. This is because of difficulty experienced during inflation to get the standard tube to fill the well without undue stretch; hence, a tube for drop-center requirements is 3 or 4 in. shorter than the standard tube so that it normally will inflate the well portion first and the tire last, that is, so that the inflated tube rests in the well.

Fig. 13 shows a 4.40-in. balloon-tire mounted on an English rim and also a similar tire mounted on an American rim. It will be noticed that the well of the English rim is much more shallow than that of the American rim; consequently, a smaller tube can be used with the English rim and less difficulty will be experienced in getting it to fill the well. Therefore, with the rim of English section, it is apparent that the tube problem is much simpler than that on the American rim and nearer to the conditions obtained in the standard flat-base rim. It will also be noted that the bead width of the English rim is 7/16 in. as compared with the 9/16 in. of the American rim-section. This allows the English rim to be built 2½ in. wide; whereas, the standard American flat-base rim is 2 5/16 in. wide and the American drop-base rim is 2¾ in. wide. The depth of the well on the English rim is 0.40625 in. and the height on the rim bead is 0.68750 in. The American rim for the same tire has a well depth of 0.625 in. and a bead height of 6.560 in. The ratio of well depth to bead height is therefore 59 per cent on the English rim and 94 per cent on the American rim.

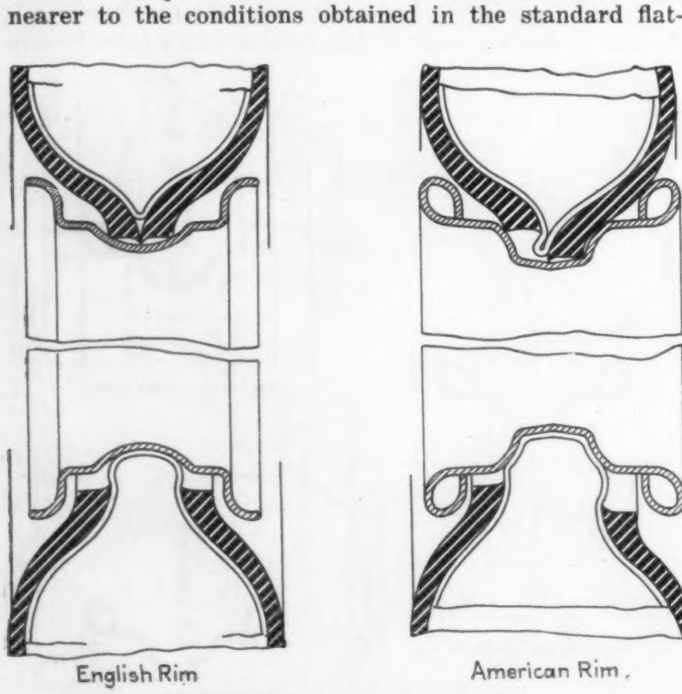


FIG. 14—TIRE POSITIONS DURING INFLATION

The Tube on the English Tire Does Not Project into the Well on the Upper Side of the Rim in the Uninflated Position While, on the American Rim, It Does So Project

Fig. 14 shows the position of the tires on their respective rims during inflation. On the English tire it is apparent that the tube does not project into the well on the upper side of the rim in the uninflated position while, on the American rim, it does so project. It also is noticeable that both beads of the English tire lay in the well while, in the American rim, there is not sufficient room for both of them and the tube is pinched between them. On the lower side of the English rim both beads hang clear of the rim seat and, inasmuch as these illustrations show the tube in process of inflation, it is easily seen that the tube has been inflated into the well on the lower side and will push the tire beads onto the tire seat on the lower half. Further, because of the favorable side-angles of the well in the rim, the tire itself will slide up onto its seat on the upper side.

On the bottom section of the American rim, the left tire-bead is in a favorable position to seat on the rim but the right-hand bead will allow the tube to be inflated so that, after the bead is pulled up to its seat, there is a considerable chance of the tube being pinched between the rim seat and the tire bead. On the upper half of the American rim, as mentioned, one of the beads lies in the well and there is also tube interference. As the inflation pressure increases, the left-hand bead probably will press onto its seat, but the pressure of the tube on the right-hand bead probably will prevent it from rising out of the well and onto its seat and, if it does slide up out of the well, there is a considerable prospect that the tube will be pinched on the corner of either upper bead.

General experience has shown that, with drop-center rims as used in this Country for experimental purposes, tube pinching does occur and that the tire cannot be depended upon to seat properly without turning the wheel and jarring the tire into place as the tube is inflated, thus equalizing the discrepancy mentioned. One American automobile builder has put out a number of cars with 4.40-in. balloon-tires on drop-center rims, and this last problem has been so prevalent that the sides of the tire have been marked with a red line just above the edge of the rim bead so that, when the tire is properly seated on the rim, it will be evident because the red line will be concentric with the rim.

Another point in connection with drop-center tires which has been commented upon unfavorably, although it is not vital, is that of the likelihood that a tire will come off the rim after a blow-out. This rarely happens on a standard flat-base split-rim, but it does happen on the flat-base rim equipped with a side ring because the side ring itself often comes off and causes very serious results. The early English tires for drop-center rims were equipped with a catch bead or rubber lip which snapped over the outside of the rim, but this has since been abandoned for numerous reasons. It adds to the tire cost, tends to limit the flexibility of the tire at the bead, and sand and dirt get under it and cause chafing. Obviously, a tire cannot come off the split rim if the tire

fits the rim base but, on the drop-center type where the bead can slip into the well, centrifugal force on the opposite side will throw the bead up and allow it to go over the edge of the rim. This can and does happen in the American rim on account of the deep well, but the condition seems to be satisfactory on the English rim on account of the shallow well. A couple of small rim tools are used with the English rim so that the bead can be pried off the rim at one point on the bead, but the tire will not come off normally when punctured.

It seems from the foregoing remarks that the problems of tube pinching, tire seating and the liability of the tire coming off the rim when punctured are inter-related. The English engineers have nearly solved these difficulties by the use of shallower wells, more suitable well-angles, narrower tire-beads, and a tube construction which is nearer to standard practice. From the standpoint of adoption, it appears that the urge for this development must come from the car builder, as the benefits to be gained are in car performance. It is also evident that additional research is needed to make this rim satisfactory.

B. J. LEMON:—Some information that I have received recently from Great Britain reads as follows:

I think it might be advisable again to emphasize the importance of a new shallow-well rim for use with the 29 x 4.40-in tire. This new rim is not yet included in standard production but it has behaved so well under test that British manufacturers are willing to adopt it as a standard if an agreement can be reached for international standardization along the lines of this section. It is said that the American manufacturer favors somewhat wider rims and it should be noted that the width of this new section is slightly greater than the standard. It is claimed that, because of its general proportions, it is practically impossible for anyone fitting a tire to inflate it and leave the tire in any other than its correct seating position all the way around the rim. The reduced well-depth also provides a considerable economy in tube cost because a normal round-section tube can be used safely. The attachment and detachment is fairly easy and can be performed with the aid of one or two small levers.

CHAIRMAN C. M. MANLY[17]:—Does not the extra width on this drop-center rim produce a noticeable effect in regard to lateral stability?

MR. LEMON:—That point has been studied intensively in Great Britain, where they have worked on this problem longer than we have. It apparently has some effect, but they say that an increase of air pressure of 2 lb. per sq. in. will give the same stability as a much wider rim. It is a question as to whether we want a wider rim having greater weight or an increase of air pressure of 1 or 2 lb. per sq. in. to maintain stability.

[17] M.S.A.E.—Consulting engineer, Manly & Veal, New York.

MOTOR-FUEL SUPPLY ADEQUATE

LOOKING ahead, over the next decade, or the next generation, or the next century, I personally continue to see adequate supplies of motor fuel for generation after generation. I am one of a number in the petroleum industry who hold that our supply of well oil itself will continue adequately to meet the essential requirements of mean motor fuel and lubricants of our Nation for generations. I believe it will also remain the cheapest source of these products. But, if I prove to be mistaken, if well oil fails in the future to constitute an ample supply, or if it becomes an expensive supply, we are certainly already in position to rest assured that we can make a motor fuel similar to gasoline from raw materials, such as oil shale and coal, which are available to us in almost unlimited quantities. Oil shale is already a thoroughly established source of motor fuel and lubricants. In Germany brown coal has also been processed to yield a motor fuel on a large scale. It is my firm conviction that these practices will be placed on sound commercial bases just as soon as gasoline prices advance to any point materially higher than our prevailing average at present.

MOTOR FUEL SUPPLIES SURE

If gasoline from well oil becomes expensive this substitute fuel will come into use. No possibility of our supply failing exists. The question is only when and at what prices other fuels than gasoline from well oil will be utilized. Well-oil supplies cannot fail suddenly. As yet, of course, the whole tendency is toward increase, but when the supply does begin to dwindle, as eventually it must, the decrease will be extremely gradual, and at that time, equally gradually, other motor fuels will commence to be produced and to be placed on the market to supplement and eventually supplant the gasoline supply.

Do not concern yourself, therefore, lest our motor fuel be exhausted. Such a contingency is not within the realm of probabilities. Do not even be alarmed lest the price go up, because motor fuel makes every promise that it will never permanently become a much larger part of the cost of motoring than it is today.—W. S. Farish, president of the American Petroleum Institute.

FUTURE MOTOR-FUEL SUPPLY

BEN E. LINDSLY, of the Bureau of Mines Experiment Station at Bartlesville, Okla., is confident that exploration, better discoveries, better utilization, and deeper drilling would furnish enough oil to meet all requirements for at least 25 or 50 years, if it could be extracted in that time. But as a practical matter, this will not be possible. Within that period times of shortage will occur when oil from shales will be needed to supplement the oil from wells.

Meantime, Federal and State governments and the industry are cooperating in an astonishing range of investigations and studies. The State of Oklahoma makes an appropriation to help maintain the Bartlesville Experiment Station, while the oil companies everywhere assist its work. The Station is carrying on investigations looking to larger recoveries, prevention of wastes, protection against fire, and effective utilization of lower grades of distillates.

Interest, when aroused at the right time, is the best insurance against disaster. The President and the Federal Oil Conservation Board have done what was needed, at the right time. The Country, the oil industry and the motor-car builders are forewarned. They will be forearmed.—H. H. Hill, chief petroleum engineer of the Bureau of Mines.

The Maintenance of a Motorcoach Fleet

By EDWARD WOTTON[1] MILWAUKEE SECTION PAPER

ABSTRACT

IN three allied motorcoach companies operating in New York City, St. Louis and Chicago, approximately 1100 coaches are in service, which travel about 40,000,000 miles per year. In Chicago, about 420 coaches render service to the three main divisions of the city.

A description is given of the system of inspection and the methods of maintaining the coaches that are in vogue in Chicago. The duties of the garage superintendents are first outlined and the manner of recording the gasoline and oil consumption and the coach mileages is explained.

All coaches are held for inspection after completing 2500 miles of operation; consequently, four coaches are inspected per day. All units are examined and any that are considered incapable of completing an additional 2500 miles of service are removed and replaced. Stock is maintained on a maximum and minimum basis. Emphasis is placed on the gasoline and oil consumption of each coach, for this is said to indicate the condition of the vehicle. All coaches go through the central repair-shops once a year, the major part of the work being the replacement of panels and repainting.

In designing the garages, 350 sq. ft. is allowed per coach and 1 inspection pit is built for every 20 coaches; in the central shops, about 100 sq. ft. per coach is considered sufficient. About 100 men are employed in the central shops, or about 1 man for every 4 coaches.

Maintenance costs are classified under the two main headings: material and labor. Daily material reports are kept on a cumulative basis so that the total cost to date can be compared with that of the previous month. A weekly personnel report shows the labor cost for the week.

TWENTY years ago this November we put the first double-deck motorcoach into operation in New York City. This coach was run for approximately 6 months as an experiment and, in July, 1907, 15 more were put into operation. The adoption of the motorcoach, however, has made very slow progress and only within the last 4 or 5 years has it been considered a serious factor in passenger transportation. At present, our three allied companies, in New York City, St. Louis and Chicago, are operating approximately 1100 coaches, which travel about 40,000,000 miles a year. Although these companies are operated independently, we are endeavoring to standardize their methods of maintenance. I will give a general outline of the method used in Chicago.

We operate approximately 420 coaches, which are housed in four garages. All major repairs and painting are performed in a central repair-shop.

We will first consider the duties of the garage superintendents, who are held responsible for the mechanical condition of the equipment in their respective divisions. They report directly to the superintendent of equipment, so that the divisions are on a competitive basis. Each division superintendent has a clerk who takes care of ready reference records of the equipment, the gasoline and oil consumption and averages, and the individual coach mileages. All reports are sent directly to the chief equipment clerk, who examines and files them in proper order and, at the end of the month, makes up various kinds of comparative statements.

[1] M.S.A.E.—Superintendent of equipment, Chicago Motor Coach Co., Chicago.

PERIODS OF OPERATION

All coaches are held for inspection after completing 2500 miles of operation. A garage operating 100 coaches represents approximately 10,000 miles, or 4 inspections, per day. Two of these inspections are held during the morning rush, the other 98 coaches being scheduled for service. All inspections are completed and the coaches scheduled for service during the afternoon rush.

During these inspections, the units are examined on a regular routine system and, if any are found defective or are not considered capable of running until the next inspection, they are removed and replaced with overhauled ones, the defective units being returned to the central shops for repair. The mechanics making the inspection have a complete report of the performance of the coach since its last inspection, consequently, if any unit has been giving trouble, they are aided by this report in forming their conclusions. A complete record of all units is kept at the central repair-shops, unit-change slips being sent through for all units removed from coaches.

Each garage has an auxiliary stockroom, which is supplied daily from the central stores. All parts are carried on a maximum and minimum basis, but not in large quantities. The minimum number of small parts to be carried is placed in a small canvas bag, so that, if the storekeeper is obliged to take parts from the bag, he knows that he is down to the minimum and should order replacements immediately. The minimum number of large parts is tagged.

The inspection system of coaches, like that of railroad equipment, is very important. It enables us to make up an estimate of personnel based on an estimate of mileage supplied by the transportation department, which gives a predetermined payroll. It assures that every man employed will have something definite to do. He is occupied at all times in carrying out work that avoids trouble, rather than waiting for trouble to develop in order to remedy it.

FAILURE OF MINOR PARTS

Failures, in most cases, are due to some apparently unimportant part's failing first. Certain parts deteriorate or wear, such, for example, as radiator hose. During inspection, the man who is accustomed to doing this work can tell very quickly whether a connection is capable of running another 2500 miles. If he is in doubt, he removes the part and replaces it with a new one, at a cost of between 40 and 50 cents. If the old one were allowed to run until it gave out on the road, it would probably mean a ruined engine. Many similar examples could be cited.

Inspection assures that all parts, such as universal-joints, wheel-bearings, and the like, are lubricated at regular and proper intervals. We do not neglect the inspection of a vehicle that happens to be running per-

fectly to relieve something else that might be tied up, for this would finally result in trouble.

When a coach is held for inspection, the oil and gas averages are checked, for they indicate the mechanical condition of the vehicle. Each division keeps on file a daily record of the gasoline and oil consumption of every coach, also the miles covered per gallon of fuel. Fuel and lubrication costs are among the highest items and should be watched constantly, coaches with low averages being given special attention.

Garage Space Allowed for Inspection

When considering garage construction, we allow 350 sq. ft. per coach and build 1 inspection pit for every 20 coaches. We are in favor, when possible, of using the indirect-heating system, which has the advantage of supplying a certain amount of circulation that can be used for ventilation in the summer, and also of supplying heat beneath the engines, where it is most beneficial in extremely cold weather.

The central repair-shops are entirely self-contained and are connected directly with the purchasing department and general stores. In these shops are maintained all units, such as engines, clutches, transmissions, rear axles, and the like. Schedules are arranged so that all coaches go through the shop once a year, the major work being replacement of panels and painting. Very little mechanical work is done, as the inspection system at the garages takes care of such work. This is much more economical than the old system of completely dismantling the chassis once a year, since no two units have the same length of life. By using our present system we obtain the maximum mileage from each unit before it is repaired.

Approximately 100 men are employed at the central repair-shops, which is about 1 man for every 4 coaches. In figuring the space for the central shops, we have found that an allowance of about 100 sq. ft. per coach is sufficient.

Cost of Maintenance

It is very important to have accurate information on the cost of maintenance and it is watched closely. Roughly speaking, maintenance costs can be classified under two main headings: labor and material. The stores department issues a daily report showing the amount of material issued under the various material classifications, of which there are 34. As this report is made up on a cumulative basis, we always have accurate information as to just how much the material has cost to date, and can compare this cost with the cost of the previous month.

The accounting department issues a weekly personnel establishment report, which shows the labor cost for the week. We have established a figure for this cost, and can readily determine when the cost of labor is higher than it should be.

Labor and material, however, are used in performing different kinds of work, and it is well to know just how much each kind is costing. The accounting department makes a monthly report of the expense accounts, which enables us to determine whether a certain kind of work is costing too much, also the relative costs of the various items entering into the maintenance of the equipment.

Method of Making Reports

A complete set of forms is used by the mechanical department, some of which I will briefly describe. A coach report card is placed on the coach at night by the same man who puts in the gasoline, at the same time that he takes out the old card. The driver notes on the card any defects or O.K.'s it.

The information on the cards is then transferred to work-sheets. Separate sheets are used for the different kinds of work, all engine and ignition work being posted on one sheet, brake work on another sheet, and so on. The next day the superintendent's clerk goes over the cards and enters the information on a general-inspection sheet. This is done daily until the coach comes in for general inspection. The general-inspection sheet is then removed from the book and put on the coach.

Mechanics look over the reports on the general-inspection sheet. If any particular unit has been giving trouble, it is given special attention. Down the side of the sheet is the regular routine of inspection that must be performed. Each mechanic signs his number after he has made the inspection called for.

In order to keep the unit records straight, all unit numbers and the average gas and oil consumption are noted on the general-inspection sheet. These sheets are then brought to my office, where they are examined and filed in numerical order.

A material-issue sheet on which are 34 classifications is made out daily by the stores department. Each requisition goes to the stockroom and is checked with its classification, so that at the end of the month we can tell exactly what any particular part of the coach costs. We divide the chassis into front axle, rear axle, clutch, engine, ignition, transmission, front and rear springs, gasoline system and miscellaneous chassis parts. Under lighting, we include generators and regulators, batteries and lamps. Under bodies, we have bar and sheet steel, lumber, paint and varnish, glass, seats, fire-extinguishers, and miscellaneous body parts. Material not used in maintaining coaches is listed under a general heading, which includes shop tools and machinery, shop expense, service cars, snow equipment and supplies, cleaning material, stationery, transportation, experimental, and buildings and fixtures.

We make up a budget at the beginning of the year that includes the amount of money we intend to spend each month. We have a payroll, of course, and know what that amounts to per day. We get the material cost by the day and know exactly where we shall come out.

Last March we analyzed a certain figure. By the middle of the month we found that it was running high, so we shut down the central shop for two Saturdays. Instead of laying off 5 or 6 men, we let all the men lose 4 or 5 hr. That helped us to keep within our budget.

Mileage of Unit Parts

When a unit is changed, a report is made on a unit-change slip. From this slip the change is noted on the coach-record sheet, so that we can tell at any time the different units that make up any particular coach, and can get the mileage of the unit by getting the mileage of the coach while the unit was on it. We also keep a card for every unit that shows the complete performance of the unit from the date it was put into service, the coaches on which it was used, the dates on and off, and the causes of removal.

We make an estimate of the payroll in advance and the actual payroll is filled in each week by the payroll department on a personnel-establishment sheet, which shows whether we are under or overestimating. We have a certain provision in which the work changes entirely with the mileage; the more mileage, the more inspections. We have one group of men on a mileage basis and another group on a coach basis. It does not matter whether the

coach runs 50 or 100 miles, it still must be washed, cleaned, oiled and gassed.

Figures for the week of April 17, 1926, when we estimated that we would work 9925 hr., show that we actually worked 9348 hr. We have no standard rates in any of the shops. We place a man at work on the floor for a while, then transfer him to greasing, then to taking off wheels, then to brakes, and gradually work him into the more skilled positions. We develop practically all our men from the ranks in that way.

THE DISCUSSION

E. A. COUSINS[2]:—Do you depend upon the drivers to report defects?

EDWARD WOTTON:—Yes. The starting-point of maintenance is generally the report of the driver.

F. M. YOUNG[3]:—Was the budget you spoke of, for maintenance?

MR. WOTTON:—Yes.

MR. YOUNG:—How can you shut down, if you have to keep the coaches running?

MR. WOTTON:—The coaches keep going. I referred only to the central repair-shops, which were up to schedule.

MR. YOUNG:—Do you make the men work faster to keep the coaches going?

MR. WOTTON:—No. We arrange to put through a certain number of coaches per month and, to do so, we figure a certain payroll. In this case, the shops were in advance of the schedule.

CHAIRMAN G. W. SMITH[4]:—Are we to understand that your work is largely preventive rather than actual repairs of breakdowns?

MR. WOTTON:—Yes.

J. H. LUCAS[5]:Do you keep the accounts by individual coaches?

MR. WOTTON:—No; we do not try to keep individual coach accounts. They are unit accounts. What we wish to find out is what the various units cost.

MR. LUCAS:—If they were costing too much, you would change them. On the other hand, you would have another kind of record, one of failures, that would really be more important than one of cost; it should go hand in hand with cost.

MR. WOTTON:—If the cost is low, the troubles are few. Unit changes are a good record of failures, however. Our monthly analysis of road delays shows the units that fail in service on the road.

MR. LUCAS:—Do you hire experienced mechanics?

MR. WOTTON:—Very rarely.

MR. LUCAS:—Do you have apprentices?

MR. WOTTON:—That is what the junior men really are.

MR. LUCAS:—What is the maximum rate you pay to first-class mechanics?

MR. WOTTON:—$0.85.

MR. LUCAS:—That compares closely with what we pay.

E. J. STONE[6]:—You use what is known as the "unit overhaul" system, do you not? By that I mean the unit replacement of engines, clutches and transmissions.

MR. WOTTON:—Yes.

MR. STONE:—Will you explain that system so that we shall get a clear idea of what unit overhaul means?

MR. WOTTON:—By the unit overhaul system we mean that any unit can be removed when defective and can be overhauled in the central repair-shop, independently of the coach. This has the advantage that it holds the coach out of service only for the length of time required to change the unit.

CHAIRMAN SMITH:—On your sheet the engine, the ignition and the clutch are the high points of maintenance cost. I can understand that being the case with the clutch, but I have always understood that the ignition system was fairly dependable. Is that due to lack of good design?

MR. WOTTON:—The item referred to includes both the engine and the ignition. The ignition, however, is maintained on a contract basis.

MR. STONE:—Have you any fixed period of mileage or can you determine in other ways when an engine or a transmission needs overhauling?

MR. WOTTON:—We go by performance and inspection.

MR. STONE:—How do you keep track of the performance?

MR. WOTTON:—By the gas and oil average and the drivers' reports; also by the noise, and the like, that mechanics discover during inspection.

MR. STONE:—What percentage of decrease in fuel economy would necessitate overhaul?

MR. WOTTON:—If there is a drop of 0.5 mile per gal. below the general average, and the engine cannot be brought up to standard.

MR. STONE:—What percentage of the average mileage would that be?

MR. WOTTON:—We average about 5 miles per gal., so, this would represent about a 10-per cent drop. Gasoline averages are very peculiar. A group of coaches doing 50 miles a day gives one average; another group doing 75 miles a day gives a better average; a group that runs as high as 175 miles a day is the best of all.

MR. COUSINS:—Under what classification on this material sheet are the tires?

MR. WOTTON:—They are on a mileage contract.

J. A. C. WARNER[7]:—How do you handle the lubrication, that is, the replacement of crankcase oil? Is that done on a mileage or a viscosity basis?

MR. WOTTON:—We change the crankcase oil every 5000 miles.

MR. WARNER:—How does it vary between winter and summer?

MR. WOTTON:—Our oil does not suffer much from the cold because, once the engine has started, it is kept running until the coach returns to the garage.

CHAIRMAN SMITH:—Do you use any auxiliary apparatus, such as radiator shutters or thermostats?

MR. WOTTON:—We use a radiator cover made of leather that has about 5 slots and can be adjusted by the driver.

MR. WARNER:—How do you handle the matter of repainting? Is that done on a periodic basis?

MR. WOTTON:—It is done yearly.

CHAIRMAN SMITH:—Do you use Duco or one of the other nitrocellulose finishes?

MR. WOTTON:—No. Our shop is not suitable for them. I do not know whether they would be economical. At present we have carpenters, electricians, and men cleaning the seats and the ceilings, all working in one operation.

MR. WARNER:—Do you take the coach completely out of service during that time?

MR. WOTTON:—Yes. It is supposed to stay in the shop an average of about 7 days.

[2] M.S.A.E.—Sales engineer, Hyatt Roller Bearing Co., Detroit.

[3] M.S.A.E. — Vice-president, general manager, Racine Radiator Co., Racine, Wis.

[4] M.S.A.E.—Chief engineer, Nash Motors Co., Milwaukee.

[5] M.S.A.E.—Superintendent of rolling stock, Milwaukee Electric Railway & Light Co., Milwaukee.

[6] District representative, Yellow Coach Mfg. Co., Chicago.

[7] M.S.A.E.—Manager, meetings and sections department, Society of Automotive Engineers, Inc., New York City.

CHAIRMAN SMITH:—Have you replaced all your poppet-valve engines with Knight engines?

MR. WOTTON:—We have not used poppet-valve engines for some time.

MR. LUCAS:—Do you use pneumatic tires on the front wheels?

MR. WOTTON:—No.

MR. LUCAS:—Then you have not had experience with hot weather troubles from pneumatic tires?

MR. WOTTON:—No.

CHAIRMAN SMITH:—What is your tire equipment, cushion or solid?

MR. WOTTON:—It is a special design. Our solid tires are about 4.5 in. high from the base. The average standard is about 2.5 in. We take them off when they wear down to 2 in.

R. C. MALEY[8]:—What do you do with regard to the inspection of transmissions? Do you omit it entirely?

MR. WOTTON:—No. We go by the noise. We watch for any discoloration of the grease in the transmission and the rear axles; when we see discoloration, we pay special attention to those units.

CHAIRMAN SMITH:—In what way do the transmission gears depreciate? Do they step off from the side on which they engage or do they pit on the face?

MR. WOTTON:—We use a constant-mesh type of transmission, with the drilled type of clutch. With the old-style dog clutch, the driver frequently dropped back into second speed while going 25 m.p.h. This he cannot do with the drilled type, without using judgment.

CHAIRMAN SMITH:—Do the gear faces fail or rub off?

MR. WOTTON:—On the old type, the edge wore off the dog, but the drilled type gives satisfaction.

MR. MALEY:—Do you find many failures that develop shortly after inspection? I suppose there is a normal amount of noise. Do you find any gears that give out, teeth that break off, or things of that kind?

MR. WOTTON:—Very seldom. In all our transmissions we use gears of different width in accordance with the work that they are expected to do. First-speed gears, being used least, have the narrowest face; second-speed gears, being used more, have a wider face.

MR. MALEY:—How about connecting-rod bearings and parts of that kind?

MR. WOTTON:—We go by the sound.

MR. MALEY:—Do the connecting-rods give out?

MR. WOTTON:—Not very often.

MR. STONE:—What is your average mileage per road failure?

MR. WOTTON:—About 27,000 vehicle-miles per mechanical delay.

MR. STONE:—What is included in failures?

MR. WOTTON:—If a driver stalls the engine and cannot crank it fast enough, we call it a delay. If the mechanic cannot find that mechanical trouble exists, we charge it to transportation.

CHAIRMAN SMITH:—What is the nature of the depreciation of the engine bearings? Is it the pounding out or the wear?

MR. WOTTON:—Wear, mostly. Since we have used a babbitted connecting-rod we have had very little pounding. We use a 3-in. crankshaft.

CHAIRMAN SMITH:—Do you use oil-filters and air-filters?

MR. WOTTON:—We have not found air-filters necessary in Chicago. We have some oil-filters in operation.

[8] A.S.A.E.—Manager of local sales and service, International Harvester Co. of America, Milwaukee.

MR. WARNER:—Do you dynamometer test your replacements?

MR. WOTTON:—No. All that we do is to test them for oil leaks, oil-pressure, and such things.

MR. YOUNG:—You said that you change the crankcase oil every 5000 miles. During that time, you have undoubtedly replaced a large part of the oil. What percentage is necessary to refill the engine each day? If the crankcase holds 3 gal., you probably put in a quart or so, do you not?

MR. WOTTON:—We check the oil every night. If 2 qt. is not required, we do not put in any.

MR. YOUNG:—What is the total capacity of the system?

MR. WOTTON:—About 11 qt.

MR. YOUNG:—Filtered?

MR. WOTTON:—With a few exceptions, only through a strainer.

MR. YOUNG:—After a long hot run, is it not pretty low?

MR. WOTTON:—No; because the general average is a little better than 200 miles per gal. We fill up the crankcase every third night. We check it every night, but all crankcases are filled up to the level every third night. That gives an easy way of checking up the bad coaches.

CHAIRMAN SMITH:—What is the nature of this oil?

MR. WOTTON:—We use a viscosity of from 74 to 76 at 210 deg. fahr. and recommend a fairly heavy oil because, if we begin with fairly high viscosity, it will stand slight dilution in the crankcase without causing the viscosity to become too low.

MR. COUSINS:—What is the average miles per coach per day?

MR. WOTTON:—The general average is about 100. Some do 30 miles; others, 175 miles.

MR. MALEY:—You said that you have an inspection once every 2500 miles. How often do you grease the coach?

MR. WOTTON:—We grease only two points, the clutch thrust and the differential, every 500 miles.

MR. LUCAS:—That is an interesting point. We have decreased the mileage between inspection to 1000. The Committee on Motor Coach Design of the American Electric Railway Association is now considering this question and is preparing a proposal that the standard practice shall be 750 miles between inspections and lubrication. If you can run 2500 miles between inspections and get 27,000 miles per delay, you have a splendid record. Did you always have 2500 miles, or have you increased the mileage? You used to have 2000 miles.

MR. WOTTON:—We began at 1500 miles on the old type of coach. With the present type, we have very few points that must be lubricated, other than the main units, as we have no spring shackles.

MR. LUCAS:—We are still operating some coaches having spring-shackles.

CHAIRMAN SMITH:—How often are the rubber blocks renewed?

MR. WOTTON:—They last approximately from 16 to 18 months.

MR. YOUNG:—How do you know when to change them? When they begin to hammer?

MR. WOTTON:—End-play finally develops.

MR. YOUNG:—Does the driver report that?

MR. WOTTON:—No.

MR. YOUNG:—If you have end-play, you have no torque-rod?

MR. WOTTON:—No.

MR. MALEY:—Do you have any connecting-rod trouble?
MR. WOTTON:—Very little.
MR. MALEY:—Are you able to detect any slight trouble through inspection?
MR. WOTTON:—Yes. A connecting-rod bearing will knock slightly.
CHAIRMAN SMITH:—What oil pressure do you maintain in the pressure-feed system?
MR. WOTTON:—About 30 lb. at 1000 r.p.m.
A. C. WOLLENSAK[9]:—When a unit develops trouble, is it removed and repaired?
MR. WOTTON:—It is removed and sent to the central repair-shop.
MR. WOLLENSAK:—Is another unit substituted?
MR. WOTTON:—Yes.
MR. YOUNG:—How much time is required to take out an engine and install a new one?
MR. WOTTON:—Two men can change an engine in 3 hr.
MR. WOLLENSAK:—What repairs do you consider sufficiently serious to warrant changing the engine?
MR. WOTTON:—Anything that holds the coach in over its regular inspection-period. We expect every coach to be out of the garage by 4:00 p. m.
MR. STONE:—Do you take such a coach out of service?
MR. WOTTON:—Not necessarily. We take the engine out, if we have any trouble at all. It is cheaper in the long run to take out the unit than to hold the coach out of service.
MR. COUSINS:—Are brake inspections made with unusual frequency, or do they run the 2500-mile limit?
MR. WOTTON:—Brake adjustments are controlled by the driver's report.
MR. WOLLENSAK:—To what standard of performance do you work?
MR. WOTTON:—We expect every brake to be capable of skidding the wheels, if necessary.
J. B. ARMITAGE[10]:—Do you have much trouble with scored crankshafts?
MR. WOTTON:—No.
MR. ARMITAGE:—Does a driver run the coach to the garage if a connecting-rod bearing burns out?
MR. WOTTON:—Once in a while. At present, only about one connecting-rod bearing burns out per month.
MR. MALEY:—Is the inspection of cylinders, pistons and rings based on the gasoline consumption?
MR. WOTTON:—Yes; we check these with a compression-gage.
CHAIRMAN SMITH:—What is the nature of the piston-ring equipment? Do you use oil-rings?
MR. WOTTON:—We have used every type of ring that we could obtain. Oil consumption is very peculiar. One engine will run 1000 miles per gal.; another, just as good mechanically, will run 150.
CHAIRMAN SMITH:—Does the engine that runs 1000 miles show any depreciation through lack of lubrication?
MR. WOTTON:—No.
CHAIRMAN SMITH:—Some engineers maintain that it is dangerous to get too high an oil mileage.
MR. WOTTON:—At one time, we had an engine that ran about 1500 miles per gal. The general average at that time was about 150 miles per gal. Mr. Green suggested dismantling the engine to find out the reason. We did so. We rebuilt the engine and remounted it, after which it, too, ran 150 miles per gal. the same as the rest.
MR. YOUNG:—What is your theory regarding that?
MR. WOTTON:—I have none.
MR. MALEY:—Is the compression test a good criterion?
MR. WOTTON:—Yes.
MR. MALEY:—Do you test the engine after it has been on a run and is thoroughly warmed up?
MR. WOTTON:—Yes.
MR. YOUNG:—What do you do with the oil that is taken out? Is it reclaimed?
MR. WOTTON:—Not at present.
MR. YOUNG:—Do you have much reclaiming of oil?
MR. WOTTON:—No; we make it a point not to fill the crankcases of coaches that are to be held out for general inspection. At present, we run on a mileage contract. We pay so much per 1000 miles.
MR. LUCAS:—You have a mileage contract on oil and tires?
MR. WOTTON:—Yes.
MR. LUCAS:—An annual contract on magnetos?
MR. WOTTON:—Yes. There is no question that this has advantages, for it predetermines the cost.
MR. MALEY:—What do you do to the wheel-bearings? Do you have any failures from the bearings getting out of adjustment?
MR. WOTTON:—No. We adjust all the bearings in the central shop. We do not depend on an average mechanic.
MR. MALEY:—When they are loose, do you take the units out and send them back to the main shop?
MR. WOTTON:—Yes.
MR. LUCAS:—Are you having any trouble with the electric regulators? Every now and then the lights go out. Are coaches taken to the garage in Chicago when the lights fail?
MR. WOTTON:—They are run to the terminal where the coach is changed. On inspections we take out every battery and replace it with a battery that has been brought up to capacity. That has eliminated a good deal of the lighting trouble.
MR. STONE:—How about the generators? Do you also remove them?
MR. WOTTON:—We take the generators out at every inspection, irrespective of their condition.
MR. STONE:—How long a time is required to change the generators?
MR. WOTTON:—Fifteen minutes.
CHAIRMAN SMITH:—Does a coach maintain a driving light like that of a passenger car, or only sufficient lighting for the purpose?
MR. WOTTON:—We need only markers for headlights. The coach lights when on full require 22 amp.
MR. COUSINS:—At 12 volts?
MR. WOTTON:—Yes.
MR. STONE:—How large is the generator?
MT. WOTTON:—Three hundred watts.
H. L. DEBBINK[11]:—After the 2500-mile inspection, do you have inspectors from the mechanical force take the coach out for a try-out?
MR. WOTTON:—It goes from the pit into service. We run a coach up to 15 m.p.h. in the garage to test the brakes.
H. D. SEELINGER[12]:—If a coach lays over at the end of a line because of trouble, who determines whether it is in condition to be used?
MR. WOTTON:—We do not argue about it. If a driver

[9] M.S.A.E.—Chief engineer, Sterling Motor Truck Co., Milwaukee.
[10] M.S.A.E.—Chief engineer, Kearney & Trecker Co., West Allis, Wis.
[11] M.S.A.E.—Superintendent of gasoline vehicles, Milwaukee Electric Railway & Light Co., Milwaukee.
[12] Jun. S.A.E.—Equipment engineer of motor buses, Milwaukee Electric Railway & Light Co., Milwaukee.

calls in and says that a coach needs to be replaced, it is replaced.

MR. SEELINGER:—About how many change-offs do you have per 1000 miles?

MR. WOTTON:—We do not say per 1000 miles. Probably not 1 coach per 100 is changed in operation. A starter might change from one coach to another and bring in one having a slight defect.

MR. STONE:—Is that the duty of the supervisor?

MR. WOTTON:—Yes.

MR. SEELINGER:—Do you have a service car on the road at all times?

MR. WOTTON:—No. We have not used a service car for years.

MR. YOUNG:—Was the service car advertised in *Motor Age* for some other purpose? It had a great deal of publicity.

MR. WOTTON:—That car was equipped as a traveling workshop for the Yellow Truck and Coach Mfg. Co.

CHAIRMAN SMITH:—Do you salvage roller-bearings or throw them away?

MR. WOTTON:—We throw them away. I do not believe in rebuilt bearings.

MR. COUSINS:—Do you use cast-iron pistons entirely?

MR. WOTTON:—Yes.

MR. YOUNG:—In the daily inspection, is the fan-belt inspected, to see whether it is tight?

MR. WOTTON:—Not every day. That is covered by the inspection at the regular periods.

MR. YOUNG:—Does the driver report, if the radiator is over-heating?

MR. WOTTON:—We get that information from the daily reports. It is not inspected daily unless reported by the driver. When the coach is in for inspection, if the fan-belt is not fit to run 3000 miles, we simply cut off the belt and put on a new one. We would rather do that than take a chance of its failing before the next inspection.

MR. YOUNG:—What do you do when overheating occurs? Do you put on a new fan-belt?

MR. WOTTON:—It might be due to retarded ignition or some other cause.

MR. YOUNG:—Then it would require inspection by a mechanic to determine the cause?

MR. WOTTON:—Yes.

CHAIRMAN SMITH:—Does not the generator in charging the battery to full capacity during the day boil off a great deal of water?

MR. WOTTON:—We have third-brush regulation. In the summer it is set at a lower charging-rate than in the winter when the lights are used longer.

K. E. KUEHL[23]:—What course of training do the drivers receive?

MR. WOTTON:—Every man must be a conductor before he drives. He does not have to be a mechanic. After he has been a conductor for several months, he becomes familiar with the traffic regulations and the operation of the coach generally. We give him special instruction. After they have completed the course in the drivers' school of instruction, the Director of Safety gives the men, in groups of a half-dozen, a talk on how to prevent accidents.

CHAIRMAN SMITH:—Are the men paid by the trip or by the day?

MR. WOTTON:—By the hour.

CHAIRMAN SMITH:—Do you have bonuses for economy in gasoline consumption or items of that sort?

MR. WOTTON:—No. We think that is a bad policy. Our business is to carry passengers. When a bonus system is installed, an unscrupulous driver will not stop to pick up passengers. We once instituted a gasoline-consumption scheme in New York City. We had maps made, on which was plotted a highway. A driver had to get more than 6 miles per gal. to qualify for the race. If a man got 10 miles per gal., he was shifted 100 miles on the map, if he got 9 miles, he was shifted 90 miles; if he got 6.5 miles, he was shifted 65 miles, and so on. The man who won the contest averaged a little more than 11 miles per gal. When he came to a traffic stop, he would shut off the engine; the conductor would go round in front, wait until the whistle blew, then crank the engine. Sometimes on a down grade, the conductor would give the coach a little push to save cranking the engine. We found, however, that the money saved on gasoline was more than offset by complaints, because of failure to pick-up passengers.

MR. COUSINS:—Is it not the practice in New York City to coast with the engine shut off?

MR. WOTTON:—Yes; on down grades.

CHAIRMAN SMITH:—Snow on the ground makes a great difference in the gasoline consumption, does it not?

MR. WOTTON:—It makes some difference, but not so much as one would expect. Gasoline consumption does not seem to correspond to the weather. On a wet day fewer tons of passengers are carried, yet very little difference is found in the gasoline consumption.

MR. YOUNG:—In the operation of coaches sold by your company in districts in which there is alkali or hard water, do you recommend that soft water be used? If they are operated where the water is hard, do you recommend that the engine, the radiator, or the cooling-system be cleaned at any time?

MR. WOTTON:—I believe a bulletin has been published on this point by the Yellow Coach Mfg. Co.

MR. YOUNG:—In Chicago, where you have a supply of soft city water, do you ever clean out the scale, drain the system completely, and remove the rust or corrosion?

MR. WOTTON:—No; we make an examination only in case of trouble with the cooling system.

MR. YOUNG:—Do you not think it would be well to do so?

MR. WOTTON:—It might be well in some localities. When the units come into our central shops, this is taken care of.

MR. YOUNG:—Should not the cooling system and the pump be treated with some solution?

MR. WOTTON:—I have not found it necessary in Chicago.

MR. MALEY:—When you make inspections in your central station, are the coaches run while jacked up, or are they run around to listen for excessive gear noises?

MR. WOTTON:—The rear wheels are jacked up.

MR. STONE:—Do you have men who specialize in the inspection of the engine, the transmission, the rear axles, and the like, who do nothing else?

MR. WOTTON:—Yes; we have specialists on each part.

MR. YOUNG:—Do you hire special mechanics? Do the men who work on the engines or the transmissions work up from scrapers, as you mentioned?

MR. WOTTON:—They work up in the garages. They need not do mechanical work. They do only inspection work, changing units, for instance, and making minor adjustments.

MR. YOUNG:—That would not be sufficient in the cen-

[23] Ajax Motors Co., Racine, Wis.

tral repair-shop where the actual repairing is required, would it?

MR. WOTTON:—No; a man who is set to washing parts and doing jobs like that develops after a while so that he can do more important work.

MR. STONE:—Have you any averages that indicate the mileage between overhauls of the engines in your fleet?

MR. WOTTON:—Yes. The average is about 60,000 miles.

CHAIRMAN SMITH:—Do you repair the sleeves, or merely the pistons and similar equipment, after 60,000 miles?

MR. WOTTON:—Mostly the inner sleeves. An outer sleeve outwears two inner sleeves.

CHAIRMAN SMITH:—It is not necessary to dismantle the engine completely, if only the sleeves need repairing?

MR. WOTTON:—No. One great advantage of sleeve-valve engines in our class of work is that the original standards, such as those of the cylinder, sleeves, pistons and rings can be maintained. When we used poppet-valve engines, after we had run 40,000 miles, the cylinders had to be reground and oversize pistons and rings had to be used. The next time the engine came in, the same thing had to be done over again. This necessitated stocking of various parts sizes that were not interchangeable.

CHAIRMAN SMITH:—You vary the sleeve and not the pistons.

MR. WOTTON:—We do not use any oversize parts.

CHAIRMAN SMITH:—Do you change the cylinder-block?

MR. WOTTON:—No; we never have to change the cylinder-block.

CHAIRMAN SMITH:—Do you have to regrind the cylinders?

MR. WOTTON:—Not at all.

MR. STONE:—How long do the blocks last?

MR. WOTTON:—I know of some blocks that have been in service 6 or 7 years.

MR. LUCAS:—They are not worn enough so that a larger size sleeve is necessary?

MR. WOTTON:—No; the travel is only about ⅞ in.

CHAIRMAN SMITH:—Do they not cut at the edge of the ports?

MR. WOTTON:—No.

MR. KUEHL:—What is the disadvantage of oversize pistons or oversize sleeves?

MR. WOTTON:—You get so much stock on hand that you cannot use it. If you stock up on standard-size pistons, you have to carry only one size that can always be used.

MR. KUEHL:—Can you not get them rough and finish them?

MR. WOTTON:—It is not advisable to do so when it can be avoided. In our garages, any piston or sleeve that is picked up and put in will work. When there are several sizes, it is possible to become confused and use the wrong size.

MR. COUSINS:—You mean that it is cheaper to stock standard sizes?

MR. WOTTON:—Yes.

W. S. NATHAN[14]:—Is it your practice, with regard to compression, to take it with the gage? You have no starter?

MR. WOTTON:—No.

MR. NATHAN:—Do you not find a great variation depending upon the speed at which the engine is turned?

[14] Jun. S.A.E.—Service engineer, Ajax Motors Co., Racine, Wis.

MR. WOTTON:—A man accustomed to it can tell the condition of the engine almost as closely without a gage as with it.

MR. NATHAN:—One man does that work?

MR. WOTTON:—Yes.

CHAIRMAN SMITH:—You have found nothing better for the job?

MR. WOTTON:—No.

MR. NATHAN:—What are your requirements? Are you interested more in uniformity of cylinder performance, or in the maximum and minimum pressures?

MR. WOTTON:—The requirements are based entirely on the gasoline and oil average and the general performance.

MR. NATHAN:—I mean as regards compression. Do you try to get the same compression in all four cylinders, or are you interested only in the minimum figure?

MR. WOTTON:—We are interested only in the minimum figure. The coach must be in fit condition to keep on schedule.

MR. STONE:—How much experimental work do you carry on at your plant?

MR. WOTTON:—Not a great deal.

MR. STONE:—What is the nature of it? Is it more in the line of body changes?

MR. WOTTON:—Body changes, seat changes, roof construction.

MR. STONE:—Not much mechanical?

MR. WOTTON:—No. We do some, however.

MR. YOUNG:—If you intend to put on a new part that is to be interchangeable with the present part, it is sent to you for test before shipment, is it not?

MR. WOTTON:—Yes.

MR. LUCAS:—Have you tried any 6-cylinder engines?

MR. WOTTON:—Yes; we have one. We cannot increase the schedule. There is no use in increasing the scheduled speed of one coach.

CHAIRMAN SMITH:—Does the 6-cylinder give less gasoline mileage than the 4-cylinder engine?

MR. WOTTON:—Yes. It gives us an average of about 3.5 miles per gal. as a general average, against 5 for the others.

MR. LUCAS:—That does not check with our figures.

MR. STONE:—The Chicago Motor Coach Co. operates over level roads, while there are grades in New York. Your purpose in experimenting, Mr. Wotton, is merely to get reliability, is it not?

MR. WOTTON—Yes.

CHAIRMAN SMITH:—The trouble is in the distribution of the carburetion, is it not?

MR. WOTTON:—I do not think so. In the coach business there is considerable idling.

CHAIRMAN SMITH:—The four cylinder distribution is better.

MR. WOTTON:—Each cylinder takes a certain amount.

MR. LUCAS:—We have found that there is not much difference between the 6-cylinder engines and the 4's and that it is due principally to the fact that there is less gear shifting with the 6's than with the 4's.

CHAIRMAN SMITH:—The 6-cylinder engine has a little greater displacement than the 4.

MR. LUCAS:—You have a higher gear-reduction and less engine-speed for the same road speed.

MR. WOTTON:—The main thing is to get a faster schedule speed.

CHAIRMAN SMITH:—Your Chicago service is all urban, is it not?

MR. WOTTON:—Yes.

MR. STONE:—Are you testing any brakes other than manually operated brakes?

MR. WOTTON:—We have one or two vacuum booster brakes.

MR. STONE:—Have you tested Westinghouse air-brakes?

MR. WOTTON:—Yes.

MR. STONE:—On two or four wheels?

MR. WOTTON:—Two.

MR. COUSINS:—Have you any four-wheel brake equipment?

MR. WOTTON:—Not in Chicago. If four-wheel brakes are used, some kind of mechanical assistance is necessary. There is too much friction loss.

CHAIRMAN SMITH:—What happens to the spark-plugs?

MR. WOTTON:—We use about two sets of spark-plugs per engine per year.

CHAIRMAN SMITH:—Do the points burn off frequently?

MR. WOTTON:—They do sometimes. This is taken care of on inspection.

MR. WOLLENSAK:—Are they the so-called one-piece plugs?

MR. WOTTON:—We have used Bosch spark-plugs for the last 2 years. We have made two or three changes. Sometimes we use a two-piece, sometimes, a one-piece plug.

CHAIRMAN SMITH:—I suppose small-wire spark-plugs would burn out every day?

MR. WOTTON:—They would not last long in our service.

MR. MALLEY:—Are you using magneto ignition?

MR. WOTTON:—Yes.

MR. KUEHL:—When you change the crankcase oil at the end of 5000 miles, do you pull the plug or drop the base?

MR. WOTTON:—We pull the plug and clean the strainer. We do not take down the base.

MR. KUEHL:—It is not much of a job, is it?

MR. WOTTON:—Every hour counts.

MR. DEBBINK:—Do you use tire chains?

MR. WOTTON:—No.

MR. LUCAS:—You said you are not using Duco. On quick repairs, what do you use?

MR. WOTTON:—Paint.

MR. LUCAS:—How long does it take the paint to dry?

MR. WOTTON:—That depends on the location of the part. We do not always wait for the paint to dry, but let it dry in service.

CHAIRMAN SMITH:—Do you keep the coaches shiny for an advertisement, or to protect them from the weather?

MR. WOTTON:—We try to keep them attractive.

MR. DEBBINK:—How many washers do you employ per 100 coaches?

MR. WOTTON:—About 1 man to 10 coaches.

MR. STONE:—How often is each coach washed?

MR. WOTTON:—That varies with the weather.

MR. LUCAS:—Do you use a shower-bath?

MR. WOTTON:—I have not found any shower-bath or mechanical apparatus that is satisfactory.

MR. LUCAS:—We have a good one.

MR. WOTTON:—On a certain type of run it is all right, but in factory districts where there is a great deal of dirt and smoke, a brush must be used.

MR. LUCAS:—A brush is used with the shower-bath; it is very satisfactory.

CHAIRMAN SMITH:—How do you route the coaches? Do they radiate like the spokes of a wheel, or is a loop and radiating service used?

MR. WOTTON:—We have three distinct divisions: one south, one west, one north; they all are run to the Loop at the city end. Each is a service in itself.

MR. COUSINS:—Are the units interchangeable?

MR. WOTTON:—Yes.

MR. STONE:—Is each of those divisions superintended by a different individual?

MR. WOTTON:—Absolutely. They are independent.

MR. YOUNG:—Do you find much difference in the comparative results in the different divisions?

MR. WOTTON:—Not so much in the mechanical repairs as in the painting. A coach on the North Side, down by the Lake Shore, is in better condition after a year's service than one running on the West Side for 6 months. A conductor on a West Side coach looks like a coal-heaver at the end of the day, while a North Side conductor looks like an office man.

MR. STONE:—Is the rapid deterioration due to a different color of paint?

MR. WOTTON:—No. I think it is due to the difference in the conditions of the run. The West Side run is through railroad and factory districts.

MR. LUCAS:—What color is used on the West Side coaches?

MR. WOTTON:—Green.

MR. NATHAN:—Would not that run have its effect on the depreciation of the automotive parts, too? The West Side run ought to show a lower engine-mileage overhaul than the North Side run.

MR. WOTTON:—Not very much.

CHAIRMAN SMITH:—What seat-covering materials are you using?

MR. WOTTON:—We use leather on the new coaches.

MR. YOUNG:—You are speaking of your own installation in Chicago, are you not?

MR. WOTTON:—Yes. In other cities, rattan seats are used. Rattan is very durable, but it cannot be kept clean and is not so comfortable because it is not flexible.

CHAIRMAN SMITH:—Some persons apparently believe that velour will supersede leather.

MR. WOTTON:—It does not wear so well.

CHAIRMAN SMITH:—Does it get dirty?

MR. WOTTON:—The corners wear.

MR. COUSINS:—Is that because of frequent changes of passengers?

MR. WOTTON:—I do not know.

MR. LUCAS:—We have had a good deal of experience with all types of pile fabrics and with leather and rattan. For sanitary reasons, the pile fabrics, of course, require closer attention to cleaning. A vacuum-cleaner does not do the work well and pile fabric is much more difficult to clean than leather. Leather also gives a better appearance when used on flexible cushions. On city street-cars, much attention has not been paid to comfortable seats, for the passengers ride only short distances. Although spring seats are used, they are not the full-spring seats that the coaches are now using and to which even city street-cars will come in time. Interurban cars have much more comfortable seating than most coaches. In them we use genuine leather.

CHAIRMAN SMITH:—Do I understand you to imply that the passenger-mile ride on a coach is greater than on a street-car?

MR. LUCAS:—What I mean to say is that street-cars for the most part are older than motorcoaches and coach passengers have a somewhat different idea of comfort in city operation. It is true there are not the oscillations in a street-car that there are in a coach. There are not the bumps or the need of cushioned seats. Street-cars being

built this year for Grand Rapids and some other places, however, are provided with full cushioned seats.

MR. WOTTON:—Some street-cars are putting in regular chair seats.

CHAIRMAN SMITH:—Does the passenger-mile service on a coach cost more than on a street-car?

MR. LUCAS:—Yes. The maintenance cost on coach equipment is about double that of a street-car.

MR. WOTTON:—In street-car equipment, the responsibility is split in different ways. You have power engineers, track engineers, line engineers. In a coach all the responsibility is on one head.

MR. LUCAS:—That is true. On the other hand, you have a vehicle with smaller capacity. A street-car weighing about 30,000 lb. is required to carry seats for 50 passengers, whereas a coach weighing a little more than 10,000 lb. carries seats for 52 passengers. The fact is overlooked that a rush-hour capacity must be provided for everybody who wishes to hang on; that as many as 125 or 130 persons are jammed into a street-car, and must be carried. On a coach, a seat is provided for every passenger, and no one is allowed to stand.

MR. STONE:—If a seat were to be provided for every passenger on a street-car, it could not be done as cheaply as on a coach.

MR. LUCAS:—A one-third increase in the number of street-cars would be needed. The passenger specification during rush-hours allows 67 seats per 100 passengers. The seats in street-cars are not spaced as closely a those in a coach.

MR. STONE:—I have seen people packed as thickly into coaches as into street-cars.

CHAIRMAN SMITH:—When people are allowed to stand in a coach you have a liability risk that you do not have in a street-car.

MR. LUCAS:—Yes. A coach is not constructed for standees. There is not the head-room, nor the facilities for standees, that there are in street-cars.

MR. WOTTON:—To my mind, the thing that has made the coach popular is the fact that it gives service. A coach can stop where the passenger is, even if he were on the wrong side of the street. The street-car driver must motion him to the other side.

CHAIRMAN SMITH:—Do not traffic-signal lights increase the cost of operation considerably?

MR. WOTTON:—No. I should say that they have speeded up traffic in the Loop about 25 per cent.

CHAIRMAN SMITH:—That is because they are coordinated lights?

MR. WOTTON:—Yes. If you get the green light and drive at the rate of 8 m.p.h., the light will change so that you can continue going.

MR. COUSINS:—Do your drivers ever take advantage of that condition?

MR. WOTTON:—Yes.

CHAIRMAN SMITH:—Can you maintain 8 m.p.h. with a reasonable number of stops and keep pretty well up with the lights?

MR. WOTTON:—Yes. Our scheduled speed is more than 10 m.p.h.

MR. STONE:—Have the automatic signal-lights had an appreciable effect on accidents in your service?

MR. WOTTON:—The number of accidents is very much less. At the same time, we have carried on a campaign against accidents through safety committees.

MR. LUCAS:—Have you a contest on accident prevention?

MR. WOTTON:—Yes; a contest is on now; not a premium contest, simply a bronze badge contest this year; a gold one, next year. There is no money bonus.

CHAIRMAN SMITH:—Do you carry liability insurance or your own insurance?

MR. WOTTON:—We carry our own insurance. In 1925, we did not carry on this campaign. In the early part of January, we went through the records, picked out 195 drivers who had gone through the year without an accident, and gave them a banquet. They had operated a total of about 6,000,000 miles without an accident.

MR. DEBBINK:—What speedometer do you use?

MR. WOTTON:—We do not use any. We get the mileage from the daily report cards.

MR. LUCAS:—Do you ever have trouble from speeding because a driver does not know how fast he is going?

MR. WOTTON:—No; each coach line has its own schedule and its own time-points.

CHAIRMAN SMITH:—Can a man leave a certain point ahead of his schedule?

MR. WOTTON:—He is not supposed to.

MR. STONE:—What is the maximum speed?

MR. WOTTON:—Sometimes coaches touch 35 or 37 m.p.h. Our recommendation is that they shall not exceed 25 or 26 m.p.h.

Front-End-Drive Symposium

VARIED means of driving the front-end mechanisms of a motor-vehicle and for banishing gear noises were enumerated and described in detail at the meeting of the Pennsylvania Section that was held on April 13, 1926. The merits and demerits of chain drive, gear drive, gears made of composition material other than metal, and metal gears having teeth specially designed to minimize noise, were threshed out. The three papers that were presented and the discussion following their presentation are printed herewith.

DEVELOPMENT OF THE SILENT TIMING-GEAR

BY E. F. BEHNING[1]

ABSTRACT

ACCORDING to the author, gear clatter and clash caused by metal-to-metal contact develops into an annoying whir or howl at high gear-speeds, and a material was sought that is flexible and resilient enough to absorb the vibrations or change their frequency to a pitch inaudible to the average human ear. Since vibrations in the crank, the cam and the generator shafts are transmitted to the timing-gears, which run at high speeds, a material was needed that would silence the consequent noise and provide a noiseless timing-gear train. A great variety of materials was investigated and the development of laminated, phenolic, condensation products resulted; these have proved mainly suitable for timing-gear-blank stock and stock for other gears such as those suitable for crankshafts and generator shafts. A further development was that of the flexible-web cam-gear made of the composition material.

The product described is made by bonding together in laminated form various bases such as paper, linen, canvas, or sheet asbestos, depending upon the grade that is to be produced, with a synthetic, phenolic, condensation resin. This resin can be hardened, made insoluble, infusible, and chemically inert to a high degree by the application of heat. Therefore, when once so hardened, it cannot afterward be softened or dissolved. Canvas-base material is the grade used in the manufacture of timing-gears, and the paper is confined to it. The other grades of the product are made in much the same manner except that other materials such as linen and paper are substituted for the canvas.

Gears made of the composition material, will, it is claimed, obviate unsatisfactory service and noise; they admit of quantity production, reduce expensive teardowns on the production line to the minimum and assure silent, positive operation after a car is in service. Such gears can be machined with the same equipment that is used ordinarily for machining metal gears, a high peripheral speed and a coarse feed being used to obtain the best results. The peripheral speed should be about 250 ft. per min. and the lateral feed about 0.025 in. per revolution, plenty of clearance being given to the cutting-tool. Ordinary, standard hobs are used for hobbing these gears; the highest possible peripheral speed should be used, with a feed of 0.075 to 0.100 in. per revolution.

THE development of the so-called "silent gear" was brought about by the demands of engineers for the positive drive inherent in a gear installation without the clatter and clash due to metal-to-metal contact which, at high speeds, develops into an annoying whir or howl. A material was wanted that would, by its flexibility, absorb the vibrations or change the frequency to a pitch that would be inaudible to the average ear. In the automotive industry, the problem was to develop a silent timing-gear train. Due to the fact that the vibrations in the crank, the cam and the generator shafts are transmitted to the timing-gears, which run at high speeds, the vibrations under these conditions develop a most annoying noise.

To overcome this difficulty, experiments were conducted with a great variety of materials. For instance, in the early days, experimental gears were actually cut from wood. These gears operated quietly but did not wear long. It was thought that vulcanized fiber-gears would solve the problem. This is a material made of chemically hardened cellulose. It is extremely tough and strong, is unaffected by oil, hot or cold, and can be machined readily. Furthermore, it had been used for years in the manufacture of gears and pinions for industrial applications. It possessed one property, however, which made it unsuitable for engine timing-gears; that is, its ability to absorb moisture. Vulcanized-fiber gears performed satisfactorily until moisture collected in the crankcase due to condensation. This moisture penetrated the gear, making it soft and spongy, and the result was that it soon failed.

Gears made of rawhide held together with end-plates had long been used successfully in industrial applications, but they, too, were found to be unsatisfactory in engine timing-gear practice because of the fact that rawhide deteriorates in hot oil. Experiments were next conducted with gears made by compressing cotton under high pressure and holding the mass under compression between end-plates by using rivets. These gears had been successful in heavy-duty work in the industrial field. In timing-gear practice, however, they were found to be unsatisfactory. In some cases the rivets failed to hold and, in other cases, the helical teeth failed to stand the end-thrust and fractured. Furthermore, these gears were relatively heavy and developed a flywheel effect on the camshaft which for some engine designs, was undesirable.

The solution of the problem was in sight with the development of the laminated, phenolic, condensation materials such as the product "celoron," produced by the company I represent, and other similar materials. This material, made with a canvas base, possessed the properties necessary to function properly in a timing-gear train, these being:

(1) Sufficient strength
(2) Resiliency; that is, ability to absorb shocks and vibrations
(3) That it be unaffected by oil or moisture
(4) That self-supporting end-plates or shrouds are not required
(5) Its light weight, since it has about one-half the weight of aluminum

[1] A.S.A.E.—Engineer, gear division, Diamond State Fibre Co., Bridgeport, Pa.

(6) Its capability for being machined easily
(7) Silent operation

Cam, Crankshaft and Generator Gears

In the early days of the development of this material for timing-gear purposes, cam-gears were made by molding a thin rim of the composition material on to a large metal center or spider. This type of gear was fairly successful and was much superior to any of the other types mentioned. However, it was found that in some instances the rim loosened from the spider, probably due to the fact that the art of molding the blanks had not reached its present state of perfection. Furthermore, it was found that, while this type of gear successfully broke up the metal-to-metal contact in the train and gave proper results on some types of engine, on other engines it was unsatisfactory due to the rigidity of the metal spider.

The next development was the use of crank and generator gears of the composition material, meshed with a metal cam-gear. This development was really brought about by the used-car trade. Certain dealers in used cars, in seeking a means of silencing the front ends of their engines to make them more readily salable, obtained from industrial gear-cutters crank and generator gears made of our material. The results were so satisfactory that an enormous demand for this type of gear developed almost over night in the replacement trade. We believe it may be properly stated that this type of gear was the first successful non-metallic gear used in automobile timing-gear trains. As a result of its success in the replacement field, the attention of engine designers was directed to the further development of the non-metallic installation.

While the non-metallic crank and generator gears gave satisfactory results as compared with all-metal-gear practice, new standards of silence were set up and even better results were demanded by the engineers. This demand was emphasized with the development of high-speed engines, with a consequent increase in the torsional vibration of the crankshafts. It was found in some installations that the mere breaking up of the metal-to-metal contact was not sufficient to give proper results, and that the small gears were not sufficiently resilient to absorb the vibrations set up at high engine-speeds. This was particularly noticeable in the generator gear which, due to its small size, failed to dampen the vibrations properly when running at a high peripheral-speed. As a consequence, in some types of engine, these gears did not give satisfactory results. Some of the gears actually broke in service, while others failed to silence the gear-train properly. Generally speaking, this type of installation was unsatisfactory when used in high-speed engines where the generator gear is actuated by the cam-gear and the generator runs faster than engine-speed. On two gear-trains, however, where the generator is driven by the fan-belt, the installation gave satisfactory results and is used today by a number of engine builders.

Flexible-Web Cam-Gear

It was recognized that, to meet the condition created by the increased torsional vibration in high-speed engines and the inability of the generator gear of small diameter, running at high speeds, to absorb the vibrations properly, some method of counteracting the vibrations other than through the resiliency of the teeth would have to be evolved. This led to the development of the flexible-web cam-gear now in general use.

This gear is made with an outer rim of our composition material having the same thickness as that of the mating steel crank-gear and with a web of our material approximately one-third the thickness of the rim. This web possesses the capability of deflecting under shock and returning to its original shape as soon as the pressure is relieved. Shocks applied to the teeth are counteracted first by the cushioning of the teeth and, second, by the deflection of the web. On a test fixture in our laboratory, a load of 2000 lb. was applied directly to the hub of a gear-blank of this type at intervals of 3 sec. for 3 weeks. At each application of the load, the web deflected $\frac{1}{8}$ in. At the end of the test, no evidence of fatigue or loss of elasticity was noticed. It is apparent that the ability of this gear to cushion or absorb the shocks and vibrations not only tends to eliminate the noise but also reduces the wear-and-tear on the surrounding operating parts.

While, for the reasons already mentioned, some difficulty was experienced on certain types of engine where the crank and the generator type of gear were used, we know of no case where the flexible-web cam-gear has failed to give satisfactory results. One important factor in this connection has been the progress that has been made by automotive engineers in the last few years in improving the balance of the crankshafts and camshafts, and improved machining practices that result in holding center-distances within close limits. Today, with the improvements in design and in shop practice and the development of the stabilized flexible-web cam-gear, conditions were never better for the installation of a gear-driven timing-mechanism giving the assurance of silent operation and long life.

Undue Expansion Overcome

In the early days of the use of gears made of laminated, phenolic condensation material, difficulty was experienced due to the fact that the material expanded in face thickness when first brought into contact with the engine heat. This was a permanent expansion or "set" and was not due to the normal coefficient of expansion of the material. The effect of this was to change the helix angle of the teeth and cause the gears to bind or develop a cross-bearing. For a time it seemed that this tendency to swell would militate against the continued use of the material for timing-gear purposes. However, the engineers of our company were successful in developing a stabilizing process through a slow heat-treatment which eliminated this difficulty and, today, gears made of our material do not increase in face thickness more than from 0.0005 to 0.0010 in. per inch of thickness at maximum engine-temperatures.

The research in connection with this stabilizing treatment developed the fact that, when the helix angle changed materially due to swelling, the teeth of the gears made from our material wore to some extent but that, after the blanks had been stabilized and a true rolling-action was attained instead of the twisting action developed by the mating of two gears of different angles, the wear on the gear made of this material was imperceptible after many thousands of miles of service.

Description of the Gear Material

The product known as Celoron is made by bonding together in laminated form various bases such as paper, linen, canvas or sheet asbestos, depending upon the grade to be produced, with a synthetic phenolic condensation resin. This resin can be hardened, made insoluble, infusible and chemically inert to a high degree of heat while under pressure. Therefore, it will be seen that, when once so hardened, it cannot afterward be dissolved or softened. Inasmuch as the canvas-base material is

the grade used in manufacturing timing-gears, our discussion is confined to that grade.

Before the development of the flexible-web cam-gear, it was the practice to saw the blanks from sheets of this material about 36 x 42 in. in size. These sheets are made by passing the canvas through a bath of resin in liquid form. This liquid penetrates the fabric, which is then passed through a kiln and dried. The fabric is then cut to the desired size and layers are superimposed one on the other until the required thickness is reached. Then the entire mass is placed between plates in an hydraulic press and heat and pressure are applied. The pressure squeezes and holds the fabric as a dense mass while the heat completes the condensation of the resin, causing it to flow and then to harden. The result is a material made tough and strong by the canvas base and bonded together and made oil and waterproof and chemically inert by the hardened resin, which in itself possesses high mechanical strength. It is a material that can be machined readily, is self-supporting and is held together by its own inherent properties. The other grades are made in much the same manner except that linen, paper or the like, is substituted for the canvas.

Gear Manufacture

In manufacturing the flexible-web gear, individual forming dies or molds are made for each size or type of blank to be produced. Discs and rings of impregnated canvas are then assembled in the mold. The loaded molds are then placed in hydraulic presses and heat and pressure are applied. One important advantage, from a manufacturing standpoint, in the production of this type of gear-blank is the ability to apply the heat more uniformly than in the case of the large sheets, due to the relatively small size of the individual molds.

Where the design of the blank calls for a metal center or bushing to facilitate the keying of the gear on the shaft, the metal center is molded in place. This center is made with a coarse diamond knurl and no rivets or other fastening devices are necessary. In fact, the center is held so securely in the blank that a force of 10,000 lb. per sq. in. is necessary to push it out on an arbor press. This method of inserting the center is much superior to the former practice of pressing in the bushing and holding it in place by rivets. Rivets come loose and, in addition, in the former practice the press-fit in some cases set up internal stresses in the blank which caused distortion. When the center is molded-in, no internal stresses are set up.

Physical Properties

Regarding the physical properties of our product, its specific gravity is 1.35, and it has a Brinell hardness on the sides or natural face of 45.0 and on the sawed or turned face of 36.4, these values being obtained by applying standard formulas to the measurement of the impression made by a 10-mm. steel-ball held under a pressure of 500 kg. for 15 sec.

Its tensile strength is 8000 to 10,000 lb. per sq. in.; its compressive strength is 40,000 lb. per sq. in. when its laminæ are horizontal and 25,000 lb. per sq. in. when its laminæ are vertical; and its transverse strength is approximately 23,000 lb. per sq. in. with laminæ horizontal and 25,000 lb. per sq. in. with laminæ vertical.

The coefficient of expansion is 0.000017 per deg. fahr., and the modulus of elasticity, or the load per square inch divided by the elongation per inch, is approximately 1,555,000 lb. per sq. in., this being an average of 10 readings.

Its impact strength was determined by an Olsen testing-machine, using a 60-lb. pendulum and measuring the loss in foot-pounds. In making this test a sample 1 in. square in section was broken by a blow delivered 20 mm. from the point of support. A blow parallel to the plane of laminations gives values of 35 to 40-ft-lb., and a blow at right angles to the plane of laminations gives values of 47 to 50 ft-lb.

Our investigation has indicated that impact strength is a more important measure of quality than tensile strength or compressive strength. This is demonstrated by the fact that Grade 10 of this material made with a paper base has higher tensile strength and compressive strength, but a lower impact strength than the Grade F made with a canvas base. The Grade-10 material was found, however, to be entirely unsatisfactory for timing-gears.

Similarly, by the use of canvas of lighter weight than the standard used by us, the machining properties of the material are improved, but this improvement is gained at a sacrifice of impact strength.

Machining Properties

Gears made of our material can be machined with the same equipment that is ordinarily used in machining metal gears. A high peripheral speed and a coarse feed should be used to obtain the best results. In turning the blank, the peripheral speed should be about 250 ft. per min. and the lateral feed about 0.025 in. per revolution. The cutting-tool should be given plenty of clearance. If this is not done, the material, due to its resiliency, will impinge on the face of the tool and generate heat, causing the tool to dull rapidly. Gears should be hobbed at the highest possible peripheral speed and with a feed of 0.075 to 0.100 in. per revolution. Ordinary standard hobs are used.

Drills should be ground slightly off center to give greater clearance and prevent heating. They should be run at high speeds. In reaming, the best results can be had with a floating reamer. The holes may be tapped if desired.

Of utmost importance to the production department is the fact that, in machining gears made of this composition material, it is not necessary to hold the gear to the rigid limits so important in metal-gear machining-practice. We hesitate somewhat to make this statement because we do not want to create the impression that we advocate loose or careless shop-practice. These gears are not a cure-all for inaccurate machining, but it is a fact that they will admit of the minor inaccuracies that are incident to quantity production. It is the practice of some engine builders using this type of gear to assemble the metal gears and the composition-material gears from stock without testing them by matching or rolling, and the percentage of tear-downs is negligible.

It seems that a gear-train is the most positive type of valve-drive mechanism. The problem, prior to the development of the laminated, phenolic condensation gears to their present state of perfection, was to produce a train that would operate silently for many thousands of miles. Metal gears can be cut to operate silently for a limited period, but difficulties are encountered when production is placed on a volume basis. This results in costly tear-downs and unsatisfactory service and, even after the job has passed the final inspection, noise will be generated when the teeth become worn. Gears made of our material eliminate these difficulties. They admit of volume production, reduce expensive tear-downs

on the production line to the minimum and assure silent, positive operation after the car is in service.

THE DISCUSSION

E. A. CORBIN, JR.[2]:—Can bevel-gears and spiral-bevel gears be made of the composition material?

E. H. BEHNING:—I know of only one installation in which our composition material is used for bevel-gears; that is in the Austro-Daimler car. This installation requires specially constructed material, so that the laminations are turned up to be at right angles with the teeth. We have been furnishing this company with such gears for about 2 years, and the result has been very satisfactory.

H. C. GIBSON[3]:—Mr. Corbin and I have been working together on gear problems. We are working out a small train of gears for a front-end drive. It involves one pair of bevel-gears having a ratio of 2 to 1, the pinion being driven by the larger gear. If this composition material is not appreciably expansive under the influence of oil, either hot or cold, could we use that combination of bevel-gears as an oil-pump and still transmit power through them?

MR. BEHNING:—I am not sure how far we can go with our composition gears for use as oil-pump gears. I believe composition gears have never been made in spiral-bevel gears, but oil-pump gears made of our composition material ran a test equivalent to 100,000 miles and still pumped the maximum quantity of oil. The only thing in such a case to be very careful about is to allow for the coefficient of expansion of the material so that the gears will not bind when they become hot. The difference between the coefficients of expansion of the two gears must be calculated and considered in connection with the center-distance. Anywhere from 0.0007 to 0.0015 in. must be allowed; otherwise, a grinding action will result which wears the gear rapidly and cuts it badly.

TIMING-GEAR DEVELOPMENT

BY R. S. DRUMMOND[4]

ABSTRACT

AFTER outlining the present status of the forms of drive for timing-gear trains, the author describes modifications of gear design made by the company he represents to overcome noise that involve lengthening gear-teeth for a given pitch. Various modifications in this regard were made and one having 16-pitch teeth with 12-pitch length had 10,000 miles of use in fourth speed without developing excessive wear.

A further development resulting from experiments was the use of case-hardened timing-gears for motor-coach engines, such usage being thought to provide the most extreme conditions. Characteristics of so-called anti-stub gears are stated and predictions are made as to the future of timing-gear practice.

HUNDREDS of thousands of automobile engines in this Country have ordinary steel-gear drives in the front end, and there are hundreds of thousands also of chain drives and of so-called fiber gears or Celoron-type gears that run in contact with metal gears. Nearly all of the higher-priced cars have adopted chain drive. In some medium-priced cars, fiber gears are driving steel or cast-iron mating gears. In the lower-priced cars, where cost has been a considerable factor, there are numerous different gear jobs. It may appear particularly complimentary to the chain drive that it has been adopted on the high-priced cars, and, in a sense, it is, but I do not think the owners of passenger-cars care much about what is in the front end of the engine; most of them do not know until they experience trouble that is evidenced by noise, but they always learn when they have noise or the engine stops running. My car stopped several times in the last 3 years that I have owned it, due to front-end-drive trouble. My troubles were partly overcome by putting in replacement parts, but I still had as much noise as before each change.

For timing-gears, long ago, plain ordinary spur-gears were used, of 8 pitch, some of them stubbed and some of them full length, of cast iron, of soft steel and of hardened steel, and varying degrees of success attended their use. Then, periodically, they developed noise troubles, which occurs on all forms of front-end drives in periods of inaccuracies that sometimes are due to the metallurgical department, sometimes to the machine department and sometimes to the assembling.

When a man encounters gear troubles, he usually goes shifting around through all the peculiar things known about gears. The engineers shifted from spur-gears to spiral gears of the ordinary type, which has a cross-cut tooth, reducing the bearing between the teeth, bringing it down from a horizontal line to a point of continuous contact at each instant. This caused some improvement in sound, probably due to the fact that new machinery and new cutters were used. The engineers then used chain for a while, and I have known them to return to the use of spur-gears and then to return to the use of the chain, and so on, ever since motor-cars have been built.

Some years ago a certain company came to us with a problem in regard to getting a quiet high-speed drive. It happened to be an overspeed-geared transmission for a car which is standard today. The company had a large gear-driving pinion, which is not a desirable gear job from the standpoint of noise. It had also a tooth design which provided just enough contact between the teeth to let the gears mesh continuously. The gear-teeth rode up on the point of each tooth before the next one took hold, and it was not a good gear job. We lengthened the tooth; instead of using 6/8 pitch we made a full 6-pitch. The 6/8-pitch teeth the company used had a real loud squeal at 60 m.p.h., and the company told its car owners that it would like to have them use the overspeed drive only at very high speeds. We overcame much of the difficulty, so that drivers of these cars could use the overspeed drive with a step-up of 28 per cent without undue noise at a speed of more than 40 m.p.h. I have driven many hours in this company's cars without discomfort. This experience led us to consider doing something more to improve overspeed drive, which has noise features similar to those of timing-gears.

DESIGN MODIFICATION

To obtain a better tooth-action, we have lengthened out the gear-teeth for a given pitch. For example, one gear job originally was designed with 6/8 pitch; that

[2] M.S.A.E.—Secretary and chief engineer, Engineering Developments, Inc., Philadelphia.

[3] M.S.A.E.—Engineering Developments, Inc., Philadelphia.

[4] M.S.A.E.—Vice-president, Gear Grinding Machine Co., Detroit.

is, a 6-pitch tooth with an 8-pitch length, being much less length of tooth than the 6/6-pitch tooth has. Our first design was a full 6 pitch. We then extended the involute curve until we secured in our later experiments a tooth of 6 pitch with a 4-pitch length. This tooth arrangement gave a greatly improved action; it had the effect of eliminating rocking action on the points of the teeth, such as occurs in stub teeth. On 6/8 pitch the tooth action involves one pair of teeth, then, momentarily, two pairs of teeth. On 6/4 pitch the tooth action involves two pairs of teeth at all times and then, momentarily, three pairs of teeth. That one effect proved to be very desirable. We carried that work still further trying not only 6/4 but 8/6 pitch, which means 8-pitch teeth of 6-pitch length; and then still further, using 12 pitch with 9 length, and then 16 pitch with 12 length.

If you were to tell the ordinary designer of motor-cars that you would like to have him change transmission-gear pitch from 7 pitch to 16/12 pitch he would probably want you shot at sunrise, but the gears are successful and much quieter than normal gear-teeth. We have a set of such 16/12-pitch gears which have had 10,000 miles of use in fourth speed. The wear on these teeth cannot be detected except with very accurate instruments as to shape of tooth and shows a possible variation of 0.3 in. after 10,000 miles of use. The manufacturer who tested them believed these gears to be a practical solution of overspeed drive with gears. We have used this type of gear in 8/6 pitch for timing-gears, our first timing-gears being cast-iron gears with a steel pinion. We then tested the design. Then we tried it out in some boat jobs using case-hardened gears entirely.

Gears for Motorcoach Engines

The foregoing trial proved so successful that we have made over 1500 sets of case-hardened timing-gears for motorcoaches. These are made in two designs; one having six gears in the train. The development was carried on in this motorcoach-engine field because we have thought that it provides the most extreme condition for test. One feature of these motorcoach gears is interesting to note. We are using 70 teeth in the idler which drives pinions of 18 teeth on the pump and magneto shafts. That is requiring a lot of a gear job or a chain job.

Anti-Stub Gears

One of the sets of gears has 230 per cent of the necessary tooth-to-tooth contact. We have put out a number of these motorcoach timing-gears that were driven with fiber crank-gears which were also ground. The gears that were used in these motorcoaches were case-hardened and ground gears which were silent as to gear-tooth noises but had a slight tone effect like a bell, due to the material used. We use the case-hardened material because it has maximum life. We know of no product that we can produce, regardless of expense, which has the length of life of the case-hardened-gear product. These anti-stub gears have the following characteristics:

(1) Due to the finer pitch, these gears give many more impacts per revolution
(2) Due to the number of teeth in contact, the "quantity" of impact is reduced
(3) When the gear-teeth are changing from one to the other, that is, when the first is going out and the third is coming in, the second is in solid metal-to-metal contact, supporting the entire load if necessary during the change
(4) When these gears are put into use, it is normal that the tips of the teeth will not show a strong bearing. It takes about 400 miles of driving or a good heavy test on a stand to bring the bearing all the way up to the top and down to the bottom. The amount of difference between taking the bearing and not taking the bearing is only a matter of tenths of a thousandth part of an inch. The idler gears are made standard for the job and variation of centers is taken care of in even thousandths of an inch on the other gears which are provided in stock, plus 0.001, 0.002 and 0.003 in., and minus 0.001, 0.002 and 0.003 in.

Future of Metal Timing-Gears

We believe that metal timing-gears will be a necessity for motorcoaches and trucks in the future. All users of motorcoaches agree that the length of life of the front-end drive is of great importance to them. They do not want to take a chassis into the repair-shop every 5000 miles because of trouble with the timing drive. Three large motorcoach users have said to me that they would be very glad to have a drive which would only need replacing every 15,000 miles. Motorcoaches make many stops and starts and this ruins the ordinary front-end construction within 5000 miles. A flexible front-end drive will not withstand continuous stopping and starting under load.

In the future we will have a great number of all-steel gears used in motorcoaches because of the length of life of such gears. A quiet job when it leaves the factory will last practically as long as the vehicle does. Any great resonance due to the action or irregularity of the engine shows as front-end noise in all types of drive. An undue amount of whip in the crankshaft will accelerate wear, and is most disadvantageous in flexible drives.

THE DISCUSSION

B. B. Bachman[6]:—I believe that, in itself, the type of drive is not the complete answer in any particular case; the problem is a combination of many details, and a variation in any one or more of those details will give a widely different result. The limitations of materials such as the effect of moisture on fiber and the effect of oil on rawhide are perfectly definite things and have a primary effect on length of life. The subjects of tooth-contours, of accuracy in the spacing of centers, of rigidity or lack of rigidity in the supporting elements or in the shafts, or the drumming tendency in the enclosing cases are all difficulties that may overlap one another so that the characteristics of each particular symptom are masked by the complete disease.

One of the fundamental difficulties under which the whole subject of gearing has labored, in common with the difficulty experienced in the matter of such a universally mechanical thing as threads, has been the lack of development of a satisfactory and easily handled method of measurement. The subject of measuring cylindrical objects or other forms of a similar nature is relatively easy; but when we come to the measuring of all the variables that may concern and may be wrong with an involute gear, it is a very complicated problem. It seems to me that a great deal more time should be devoted by automotive engineers who are interested in gear problems toward getting a more fundamental idea of the work which has been done along that line.

Quoting from my own experience, I think that possibly the lack of specific knowledge regarding the details of gear construction itself, on the part of the individual

[6] M.S.A.E.—Engineer, Autocar Co., Ardmore, Pa.

who is engaged in general engineering, is responsible for some of our difficulties. We have been tied down by convention in the form of general information to which we have had access which dealt with the relations that exist between the center-distances of gears and the pitches of the gears as normally expressed, generally in diametral pitch. In fact, the subject of the diametral pitch of an involute gear has relatively little significance except that it is a convenient method to use as regards arithmetic. The involute pitch, the distance between the involute curves of a gear, is really the fundamental of gearing as we have come to feel about it in recent years and, with that thought in mind, many things can be done with regard to the proportioning of gears that otherwise might be lost sight of.

Mr. Drummond's description of changing the proportions of gear-teeth is extremely interesting. I do not know what the basis was upon which the original proportions of gear-teeth were established, in which the working depth of the tooth was equivalent to twice the reciprocal of the diametral pitch, but we know that this has been the accepted proportion; and then a certain company in this Country was responsible for promoting the so-called stub-tooth form in which, instead of having the working depth of the tooth twice the reciprocal of the diametral pitch, a shorter tooth than that was used. In some cases, with coarser pitches, there is only one pitch-difference and, in others, two. That was done primarily for the purpose of developing strength in gear-teeth and was initiated at a time before the character of the materials now available was known. In his work, Mr. Drummond has gone just the other way around with the idea of increasing the tooth-contact, and that opens up a field for considerable speculation as to what may evolve from such a move.

R. S. DRUMMOND:—The subject of measurement and the inspection of the product is of interest to all automotive engineers. Gear-inspection devices are in their infancy. A few years ago I made a summary of all the available inspection devices in this Country and abroad, endeavoring to secure the finest equipment for our company. I was appalled at the errors which were allowable in such devices. I found some of them made in ways that were unique but not very useful. Some devices measured something, but no one knew exactly what. I remember that one device was brought to us for acceptance in which the allowable error was 0.0030 in. and we were trying to measure 0.0005 in. But accurate measuring devices of one make or another have improved the production of gears so much that many plants now have new master-gears.

MR. BACHMAN:—Has Mr. Drummond any specific information as to the relative power that can be transmitted by two gears having the same diameter and face and composed of the same material, with normal tooth-proportions and with the "abnormal" proportions which he proposes? On account of the extreme pointedness of the tooth, will not difficulties develop in case-hardened material in getting a very brittle point due to high concentration of carbon at the end of the tooth?

MR. DRUMMOND:—As to the relative transmission of power of two gears of the same face, diameter and material under normal conditions and normal tooth-proportions, compared with the unusual tooth-proportions which I suggest, I cannot answer the question simply by saying yes or no. The length of life of a gear is an entirely different thing from the breaking load on the tooth. A gear is a good gear so long as the surface of the gear remains intact in its original form. Therefore, the factor of the breaking load is not of moment as compared with the factor of surface wear. We are dealing with something that does not have a hammer impact. It is a wearing away of the surface, the surface finally pits and is gouged out; then the tooth snaps. So far as breaking strength goes, the timing-gear of the construction I described could transmit its load without breakage of the teeth in the ordinary sense in a very narrow gear, and that is true also of transmission-gears; but when it comes to the length of life of the gear which is limited by the wearing away of the surface, the ultimate pitting and then the gouging out, it is entirely a matter of accuracy of surface condition and of the materials used. The accuracy can be attained and we are making great advances in the development of suitable material. The next step in extending the length of life of a gear is dependent on the design and on the accuracy of the machine work. I estimate that the two gears in question would have a comparative length of life of 3 to 1, compared to gears of usual types and usual methods of manufacture in automobile transmissions.

With respect to the pointedness of the teeth, we have made semi-pointed gears in quantities of 150 jobs per day for several years without having tooth-breakage. The ends of the teeth at the faces of the gears are chamfered off so that the sharp corners will not be battered up. We have found no strain in the use of the gears which affects that very sharp edge, which approximates 1/32 in. and is no doubt case-hardened throughout. We have clashed gears of this type in 12-pitch with 9-length, but have cut back the tip to get mesh on the thicker part of the tooth. We thus avoid damaging the sharp edge when the clash takes place. We have found no difficulty with thin edges if made from properly case-hardened materials. Approximately 0.006 in. is left on the tooth surfaces of the gears which is removed by the grinding process, and the outside diameter is ground with the ordinary grinding-tolerance. I think that is an important factor in their construction.

The center of these case-hardened gears is of tough steel. The core of the gear is not a separate entity from the skin surface; the microscope shows that they actually blend into each other. If the case-hardening is wrong because of poor modification of the "case" with respect to the core, the tooth-grinding process will show this in checked gears. One can develop crazy cracks all over the ground surfaces, not because the grinding has been improperly done but because the case has not been properly applied and the core has not been properly treated.

The stub-tooth was only introduced to increase the breaking strength and without due consideration of wear or accuracy or silence. The use of stub-teeth operates against silence, makes a less acceptable design and the length of life is shortened. This has been proved so well that the instances of stub-tooth design are becoming rare. Formerly, we could count on our fingers the users of full-length gear-teeth, but now the users of full-length gear-teeth are many.

CHAIRMAN C. O. GUERNSEY[7]:—Regarding variations in manufacture, what reasonable variation can be allowed on gear center-distances in some given set of typical conditions without expecting trouble with gears of the type Mr. Drummond mentions?

MR. DRUMMOND:—With regard to reasonable variation of centers and the possibility of accommodating wide ranges of center-distances, it is customary to make these gears with a backlash of 0.003 or 0.004 in. They all ex-

[7] M.S.A.E.—Chief engineer, automotive car division, J. G. Brill Co., Philadelphia.

pand when they become hot and operate very satisfactorily with a tolerance of even 0.006 to 0.008 in. if there is no unusual amount of whipping, no unusual knocking of the engines. I believe that the expansion and the contraction would not cause serious difficulties up to a tolerance of about 0.006 in. I know of jobs that are put together metal to metal when cold.

CHAIRMAN GUERNSEY:—What effect on these gears can be expected from the use of what might be called a floating bearing on the crankshaft; that is, a bearing fitted with considerable clearance and substantially held in place by oil pressure?

MR. DRUMMOND:—We have not had much experience in that respect; a great deal would depend upon how well it floats. An undue amount of floating of any gear or similar device changes the whole picture of tooth-contact and gets us far away from the exact theory of gear-contacts because the gear is out of center with a shaft. It is difficult to answer that question directly without considering every detail concerning it.

CHAIRMAN GUERNSEY:—In our class of work, considerable length of life is of prime importance and, while we do not like noisy cars, silence is perhaps secondary to length of life.

MR. DRUMMOND:—Each one of us who works on timing-gears should hold before himself as ideals the quietness demanded by the customer and the length of life without repairs or adjustments. The former is of greatest importance in passenger-car work, and the latter is of maximum importance in motorcoach and in motor-truck operation. These two fields of usefulness for timing-devices are wholly distinct. In a passenger-car, we can allow only a fraction of the noise allowable in a motorcoach or a motor-truck. On the other hand, I think we must provide a much longer life in the timing-devices on motorcoaches or motor-trucks. The demand for longer life of timing-devices on motorcoaches and motor-trucks is a present one and has a serious effect on sales.

E. W. TEMPLIN[8]:—Concerning our experience in the manufacture of motorcoaches, our two fundamental requirements are the maintenance of dependable schedules and the avoidance of service calls from the road on account of tie-ups. Our company has eliminated front-end-drive troubles by using the type of gearing that Mr. Drummond has developed. We have built, in some respects, a special vehicle for use across the Syrian desert; it is a 600-mile run during which there is no chance for service of any kind between the two terminals, Damascus and Bagdad. On a test run recently, I could hear no sound inside the vehicle due to the timing-gears; however, I could hear a very low tone when standing outside it. I have no misgiving about the good performance of the gears in the front end, or that the vehicle will be tied-up on the desert due to failure of the gears.

CHAINS FOR FRONT-END DRIVES

BY F. M. HAWLEY[9]

ABSTRACT

TOOTHED and friction-gearing are said by the author to be the two distinct classes of power transmission between two shafts, and the silent chain he describes is in the toothed-gearing class according to his statement, since it has a fixed speed-ratio and causes a bearing pressure that varies almost directly with the power transmitted. It is argued that, because of its elasticity and the peculiar method of contact with the teeth of the sprocket, the silent chain constitutes a medium that absorbs shocks and variations in angular velocity, and has a bearing action similar to that of a belt.

The improved silent chain is made of stamped, arch-shaped link-plates assembled in alternate succession and joined by pins that act as bearings. The spacing of the pins forms the "pitch" of the chain. When assembled, the chain can be considered a flexible gear or rack. The projecting teeth of the link-plates engage the sprocket-wheels over a considerable arc of the periphery of each and reduce the pressure per tooth, thus minimizing tooth-wear.

Illustrations and descriptions of the rocker-joint, the pin-type joint and the constant-pressure-angle chain are presented and, in addition, a typical layout employing the triangular and other types of drive. In conclusion, it is said that all engineering requires compromises in the design of parts and that this is true also regarding silent-chain drives. The object of their use is silence and satisfactory length of life, both of which are controlled mainly by design; hence, great care should govern the design. The successful drives are those that have been developed after careful consideration of the advantages to be gained and the pitfalls to be avoided.

[8] M.S.A.E.—Engineer, motorcoach division, Six Wheel Co., Philadelphia.

[9] M.S.A.E.—Engineer, Morse Chain Co., Detroit.

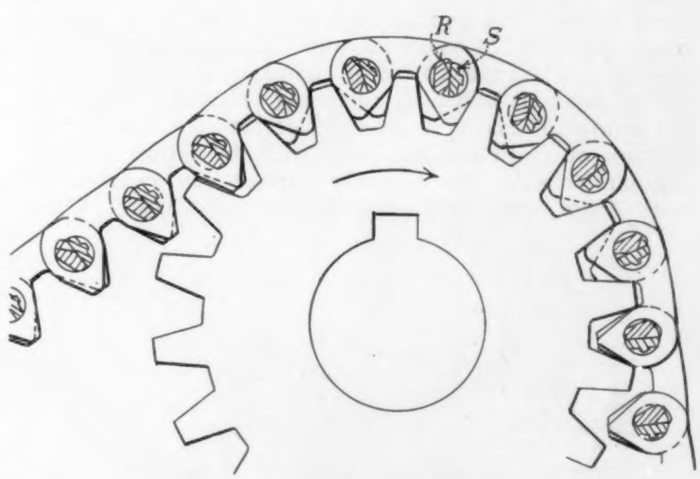


FIG. 1—THE ROCKER-JOINT

The Illustration Shows Also the So-Called "Inside Method" of Engagement. The Two Pins Form the Joint. The Pin Designated R Is the Rocker-Pin and the Other, S, Is the Seat-Pin. The Rocker-Pin Is Semi-Triangular in Shape, with the Pointed Side Bearing Against the Broad Flat Side of the Seat-Pin. Each Pin Is Fixed in the Link at the Point Nearest to the Ends of the Link and Has a Clearance at the Portion of the Hole to Which It Is Not Fixed. This Clearance Gives the Joint Freedom To Operate

AN analysis of power transmission in use between two shafts shows two distinct classes, tooth-gearing and friction-gearing. Upon examination of these two classes, we find that each possesses certain characteristics. Silent chains are classed under tooth-gearing. A silent chain has, among the characteristics of the class, the fixed speed-ratio since it is driven by the contact of the third jointed link-member with the teeth and forms a bearing pressure which varies almost directly with the power transmitted. The silent chain, through its elasticity and peculiar method of contact with

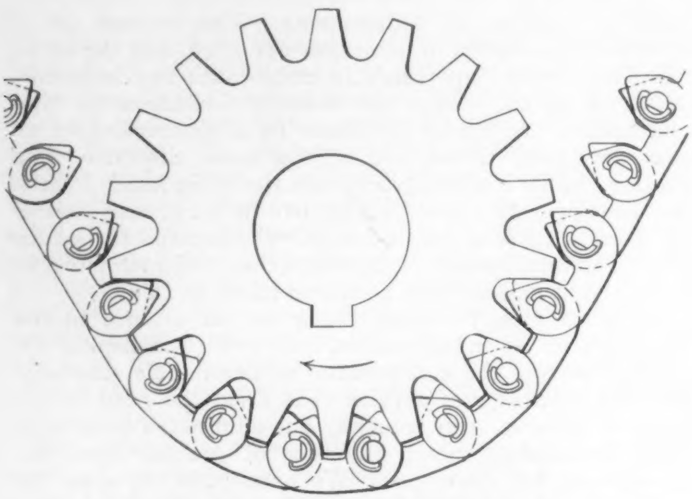


FIG. 2—THE PIN-TYPE JOINT
The Action of This Type Approaches More Nearly to That of the Ordinary Journal-Bearing. As the Change of Position of the Links Takes Place There Is a Rotating Action of the Pin Against the Bushing, or Vice Versa. The Pin Is Held Securely by Extending Through the Outside Link in a D-Shaped Hole and Being Riveted

sprocket-teeth, forms an elastic medium that absorbs the variation in angular velocity and shocks found in many gear systems and produces a bearing action similar to that of a belt. The application of silent chain to cam and accessory drives in automotive engines is therefore unique and logical.

Chain permits a reasonable range of variation of pitch-centers. It increases the possibility of more accessible location of accessories such as the generator, water-pump, distributor and such equipment, and eliminates "tear-downs" and mating such as is common in gearing where quietness is essential, and this is an economic advantage. The replacement of chains in service is accomplished without difficulty, restoring original quietness at the minimum cost. A gear drives through the sliding-friction of tooth against tooth and, to obtain uniform angular velocity, theoretically correct center-distances, tooth form and tooth spacing are necessary. These necessities call for refinements in manufacture that are difficult to maintain, and anything materially short of perfection decreases the efficiency and increases the noise of the drive.

The improved silent chain is made of stamped arched-shaped link-plates of approximately 0.050-in. thickness, in timing-chain practice, assembled in alternate succession and joined by pins which are no less than bearings. The spacing of the pins thus forms the pitch of the chain, which is now commonly found in 3/8, 4/10 and ½-in. pitch. When assembled, the chain may be considered a flexible gear or perhaps is better described as a flexible rack. The projecting tooth of the link-plates engages the sprocket-wheels over a considerable arc of their circumference; thus, the pressure per tooth is reduced to the minimum, which practically eliminates tooth wear.

The link engagement with the tooth is at a point below

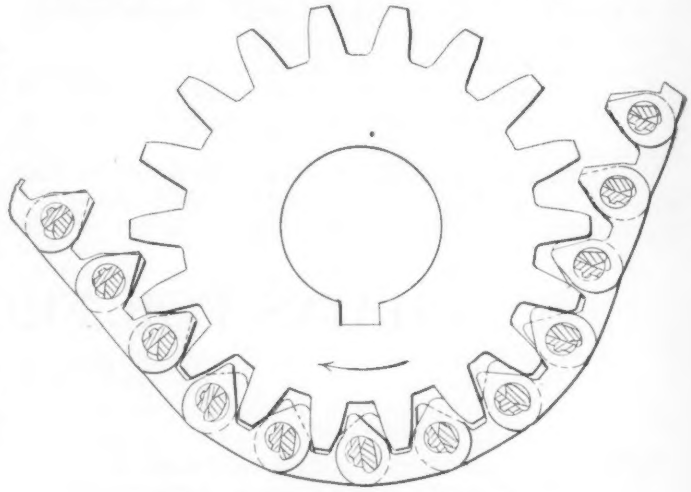


FIG. 3—CONSTANT-PRESSURE-ANGLE CHAIN
The Peculiarity of the Design Is That Each Sprocket-Tooth Contained in Wheels Up to 33 Teeth Is Beveled at the Top at an Angle with the Radial Center-Line of the Tooth That Is Constant with All Sprockets. The Engagement Is on the Inside of the Driven Sprocket Taking the Pull, and on the End When the Sprocket-Tooth Is Doing the Driving

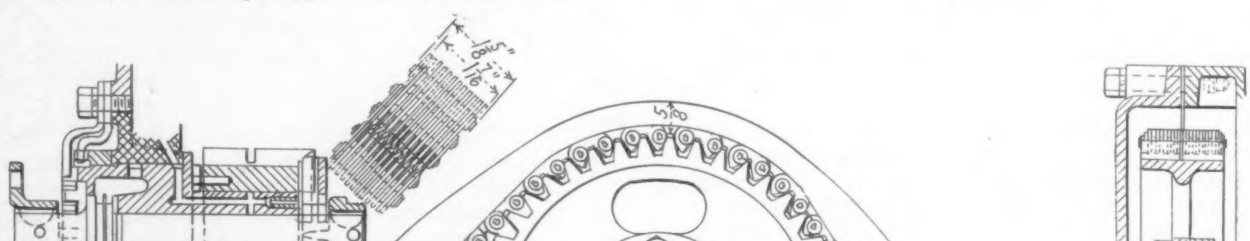


FIG. 4—TYPICAL TRIANGULAR DRIVE
In a Triangular Drive, the Position of the Camshaft Largely Determines That of the Generator or Accessory Sprocket Due to Angular Contact Necessary on the Crankshaft Sprocket, and the Fact That It Is Desirable To Relieve the Smaller Sprocket of As Much Load As Possible, and Not To Transmit Any Load Over It

the line of pull so that, in a small driving-sprocket where the angle of tooth to the line of pull may sometimes be less than the angle of pressure there exists, in addition to any centrifugal force that may be present, a lever action that tends to maintain the chain at a point higher than its proper pitch. Where there are more than 20 teeth in the sprocket, the contact with the tooth is at an angle greater than the angle of pressure, which tends to make the links slide up the teeth, and this tendency of the chain to take a larger pitch-diameter on the sprocket-wheel automatically compensates for the lengthening of the chain and maintains a correct pitch-relation between the chain and the teeth of the wheel.

Three forces are present which tend to slip the chain outwardly, requiring a definite opposing force to keep it in place. A larger part of this accumulated restraining force is represented by the frictional resistance to sliding up the teeth, which depends upon the number of links in engagement as well as upon the size of the sprocket, since the force varies with different sizes of wheel. Experiments have shown that from 5 to 30 teeth in engagement, depending on the angle of tooth to the line of pull, are sufficient to keep the link farthest away from the direction of pull from rising on the tooth with an appreciable force.

In a proper design, say with five or more teeth engaging, before leaving the driving-sprocket the chain will float between the teeth, barely touching either face, until the tension of the slack side pulls the chain against the idle or non-driving face of the teeth. The effective driving force of the chain is the difference between the tight and the slack strands, and something of the same "creep" that is found in belting takes place on the tooth face, since the chain is driving through frictional contact with many teeth.

On the driving-wheel, the first or entering link engages with the tooth when moving in nearly the same direction and speed as the sprocket, and takes up its burden of driving with little shock or noise. The first link engaging takes the largest share of the burden of driving and, due to elasticity, is longest at this point and rides highest on the sprocket-tooth.

The lengthening of the chain links due to elasticity decreases progressively around the wheel, and this permits the chain to creep down the teeth of the driving-wheel until it reaches a point at or below its pitch-diameter. Under these conditions, the chain is drawing a spiral on the sprocket, with the larger diameter on the driver at the entering tooth and on the driven at the leaving tooth, while, somewhere between, where the tension on the slack side becomes operative, the smaller diameter will be found. It is this distribution of tension necessary to drive over a large number of teeth that causes the chain to be quiet in action, since the pressure of driving is taken up gradually and without the effect of a blow.

The silent chains on the market today may be placed in two general classifications; one is the form of joint in which we find the rocker or frictionless joint, and the pin-type or friction-joint; the other is the method of tooth engagement which is defined in the so-called "end engagement" and the "inside engagement."

THE ROCKER-JOINT

Regarding the joint construction, the rocker-joint type appearing in Fig. 1 has been used very extensively in automotive timing-chain service. Morse first obtained patents upon this type of chain in 1893, and it is to the bicycle, perhaps, that thanks should be given for the impetus which led to the present development in chain

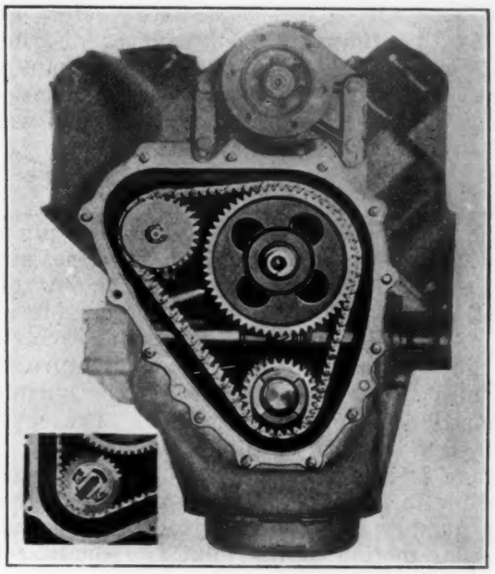


FIG. 5—"AUTOMATIC-ADJUSTMENT" TYPE OF DRIVE

This Design Is Used on a Car Equipped with the V-8 Engine. The Camshaft Only Is Driven by Chain, an Extension of Which Takes the Drive-Pulley for the Fan and Generator, with the Automatic Adjustment Used as the Third Sprocket

driving. The two pins form the joint; the pin designated as R is the rocker-pin and the other, S, is the seat-pin. The rocker-pin is semi-triangular in shape, with the pointed side bearing against the broad flat side of the seat-pin. It will be seen that each pin is fixed in the link at the point nearest to the ends of the link and has a clearance at the portion of the hole to which it is not fixed. This clearance gives the joint freedom to operate. When the chain is off the wheel, each joint has to bear the full pull. At that time there is an appreciable flat surface presented to carry the load. As soon as the chain starts to wrap itself about the wheel, it breaks

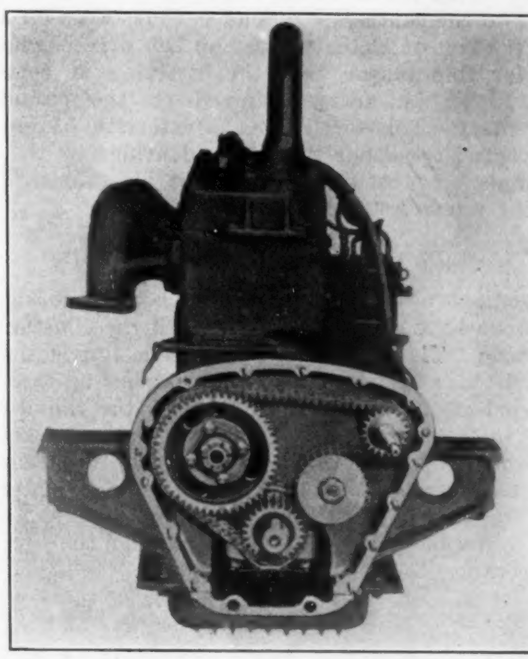


FIG. 6—TYPE OF DRIVE USING DUPLEX CHAIN

The Duplex Chain Increases the Possibilities of Chain Driving by Its Advantage in Covering Shafts of Fixed Centers or Extraordinary Design. Where the Accessory Shaft Is at the Right of the Camshaft, the Crankshaft Sprocket and Thence to the Accessory Crankshaft Sprocket and Thence To the Accessory Sprocket through the Automatic Idler

the surface bearing and immediately offers a line contact. At this time, the load becomes distributed over several joints. The links and the two pins are case-hardened so as to afford the best wearing qualities. The ends of the seat-pins are annealed, and washers are riveted over them when assembled.

PIN-TYPE JOINT

In the pin-type joint shown in Fig. 2 or pin and bushing as it may be called, the action approaches more nearly to that of the ordinary journal-bearing. As the change of position of the links takes place there is a rotating action of the pin against the bushing, or vice versa. In the pin-type joint of our own manufacture, the pin is held securely by extending through the outside link in a D-shaped hole and being riveted. The semi-bushing is retained in the adjacent link by a specially shaped hole. Advantage is held in this type of joint in the smaller pitches due to increased bearing-area. By virtue of the greater number of teeth in sprockets of smaller pitch, the angular motion of the joint is reduced, which also enhances its possibilities.

In the method of engagement, Fig. 2 shows that the link-plates are symmetrical, which also indicates that either end of the link engages the sprocket-tooth over its entire flat surface while the arch of the link clears the tooth. This means that either end of the link becomes the working face, depending upon whether driving or driven. Bearing on the sprocket-teeth will therefore show well down the face of the tooth on either side and the angle of pressure varies directly with the number of teeth.

With the inside engagement indicated in Fig. 1, it will be noted that the link is unsymmetrical and that the larger end of the chain or that portion ahead of the engaging tooth is entirely clear of the tooth-gap. The flat surface at the end of the link takes the drive load, engaging the full face of the sprocket-tooth and, when the chain pulls the sprocket, the load is taken on that portion of the radius at the inside of the link drawn concentric to the center. The load is thus taken on a fairly short portion of chain contact on the driven wheel, and it is for this reason that the direction of rotation is indicated by an arrow stamped on the outside link of the chain. This feature is characteristic of our chains and practically all our types are identified by the arrow. The angle of pressure in this type also varies directly with the number of teeth in the sprocket.

CONSTANT-PRESSURE-ANGLE CHAIN

In some types of our chain, such as that shown in Fig. 3, we have adopted what we call the constant pressure-angle. The peculiarity of this design is that each sprocket-tooth contained in wheels having up to 33 teeth is beveled at the top at an angle with the radial center-line of the tooth that is constant with all sprockets. At 33 teeth this angle coincides with the tooth face and takes the reverse direction at its apex at a greater number of teeth. The engagement is on the inside of the driven sprocket taking the pull, and on the end when the sprocket-tooth is doing the driving.

CONSTRUCTION

About the only thing that can be said of the detail construction of silent chains, that is common to all, is that the opening angle or the angle formed by the end faces of the link is the same, or approximately 60 deg. It was found that this created a desirable range in numbers of teeth in the sprockets at which ratios up to about 10 to 1 could be accomplished. The limitations are based, of course, upon the shape of the teeth. It develops from the formula, angle of pressure = 30 deg. — (360 deg. $\div N$), in which N represents the number of teeth, that at 12 teeth the angle of pressure coincides with the pitch-line, which is the minimum; and that at an infinite number of teeth, which would produce a rack, the angle of pressure becomes 30 deg., the maximum.

The tooth form becomes pointed at about 120 teeth, and dubs off as the number is increased. Hence, the range exists between 12 and 130 of a possible number of teeth. It is undesirable to accept either of these extremes for practical service, so we have chosen 14 teeth as the minimum. The top number is entirely out of our range of need in timing-chain practice, so it will not be discussed.

A noise factor is introduced in the smaller sprockets due to pressure and friction; for this reason, it is desirable to evade these smaller-tooth sprockets wherever possible. The sprocket pitch-diameter is of course developed by the chordal pitch-center and determined by the formula pitch-diameter = $P \div$ (sine 360 deg./$2N$), in which P is the pitch.

The outside diameter varies in practice. It will be seen that the chain must not strike the top of the tooth; so, the tooth height should be just enough to give the chain proper bearing-contact at the face or engaging surface of the link. The outside diameter is, therefore, roughly, the pitch-diameter. The depth of cut or tooth-gap is that required to clear the bottom points of the link.

APPLICATION

The basic principle of the silent chain thus analyzed, its application is a point of special study. Granting that sprocket-wheel sizes and positions are suitably worked out, various phenomena may affect the quietness and durability of the drive. A condition of resonance may occur, traceable perhaps to the chain, but due to elastic distortion of metal somewhere in the crankcase or chain housing. It might be called acoustics which permits certain common noises to become magnified and objectionable, and happens so either by design or methods of molding and cooling the metals. Ribbing of flat sections usually offers an effective remedy for elastic distortion or "drum" and should be observed in design.

Engine vibration sometimes causes a peculiar disturbance in chain, in the form of a "whip" in one or more strands. This produces a rough marbly noise and occurs at some particular speed or period. It has been demonstrated that these periods exist in synchronism, and affect the chain so far as it "tunes in." While the vibration may not be evident to the sense of feeling, or the speed at a point that would be called "critical," a breaking up of the forces with which the chain synchronizes always removes the whip. It points, therefore, to a torsional condition in the crankshaft and a vibration of low amplitude, doubtless aggravated by impulsive action of the camshaft due to valve-spring load. An erroneous opinion is sometimes formed regarding chain whip or periodic noises in that it is thought to be faulty chain or improperly machined sprockets, neither of which materially influences the condition.

Considering the usual range of conditions as advanced by the engine designer and to which good practice dictates in chain-drive design, its application is very broad and conveniently adapted to the standard type of poppet

or sleeve-valve engine. Gear drives are often replaced by chain with only slight changes in the front-end housing, and possibly the direction of the camshaft.

A Typical Layout

Fig. 4 exhibits a layout of a typical triangular drive of our design. It offers simple construction and is most economical from the viewpoint of chain length. In considering a triangular drive, the position of the camshaft largely determines that of the generator or accessory sprocket due to angular contact necessary on the crankshaft sprocket and the fact that it is desirable to relieve the smaller sprocket of as much load as possible, and not to transmit any load over it. This places it at the left of the camshaft, facing the front. It is important to maintain the shortest length of chain between the crankshaft and the camshaft, to decrease timing lag in the event of chain wear. In this layout, a sprocket combination of 21, 42 and 18 teeth is used, and a standard ½-in. pitch chain of the rocker-joint type, 1½ in. wide and 63 links long, one of which is the "hunting" link.

In establishing the position of the sprockets, it should be done with the idea of gaining the maximum angular contact on each; normally considered, not less than 95 deg. or five teeth. Sprocket sizes in number of teeth possess a measure of importance upon chain life and quietness. Experience has demonstrated that the chain will operate at a pitch velocity up to 2400 ft. per min. without appreciable noise, and will transmit load safely to a speed of 3600 ft. per min. without excessive wear or noise above characteristic engine noises common at this speed. Therefore, the crankshaft sprocket of 18 to 21 teeth and ½-in. pitch is logical. Satisfactory charging speed of the generator must, of course, be maintained, in which no difficulty is experienced with a ratio of 1¼ to 1, and sometimes less, to crankshaft speed. As previously indicated, it is important to maintain the sprocket size at or above 16 teeth; so, an adjustment has been agreeably reached between the electrical-equipment manufacturers and the chain manufacturers.

Chain Width, Lubrication and Adjustment

It is difficult to advance a rule regarding the selection of proper chain-width in designing a drive. We usually depend upon experience more than upon a rule. The width employed depends upon the size of the engine and the number of units driven by the chain. It is seldom less than 1¼ in. wide as used in engines up to 200-cu. in. displacement; above 200 cu. in. and up to 300 cu. in., chain will be found to be 1½ to 1¾ in. wide. Larger engines usually require still wider chains for satisfactory life.

Good lubrication on the chain is essential and should supply it with a steady flow of oil from the pressure feed. Much discussion and lengthy articles have been advanced during the last few years regarding the use of fuels and their effect upon dilution, and the chain has suffered no small part of the bad effects. Corrosion has a distinct effect upon chain, on account of the nature of its small bearings and should be protected from it.

The necessity for adjustment of the chain depends somewhat upon the type of chain and the nature of the drive. On the triangular drive, adjustment is imperative to maintain satisfactory chain life. Its purpose is mainly to take up slack due to wear, which increases materially its usable period by preventing sufficient slack

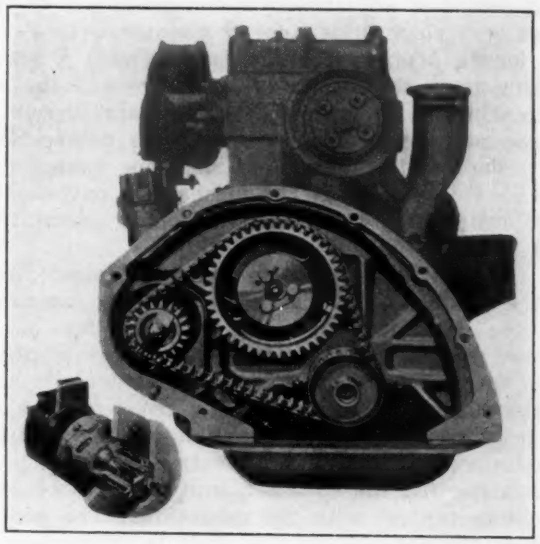


Fig. 7—Spring-Drive Generator Sprocket
The Sprocket Is Mounted on a Specially Designed Bracket Having an Extension That Forms the Bearing. The Generator Flange Attaches Flush to This Bracket and the Chain Is Manually Adjusted by Moving the Generator Outward, Pivoting on the Lower Cap-Screw

to allow jumping or slipping teeth, and the throwing of the engine out of proper timing.

Manufacturing accuracy is now possible in silent chain to fit the sprockets of a given layout exactly to the proper running condition; so, the purpose of adjustment to set the chain up correctly at initial assembly is nil. Pitch tolerances in our plant are rigidly held to 0.0005 in. per pitch. The total lengthening a chain will withstand within its usable period is approximately 0.020 in. per pitch in ½-in. pitch chain. In a 63-link chain, it will be seen that the total lengthening will be 1.26 in., or approximately 1¼ in. An adjustment of one-quarter of this amount, or 5/16 in., will consume a sufficient amount of the chain lengthening to remove one link and set the adjustment to its original position. When the sub-

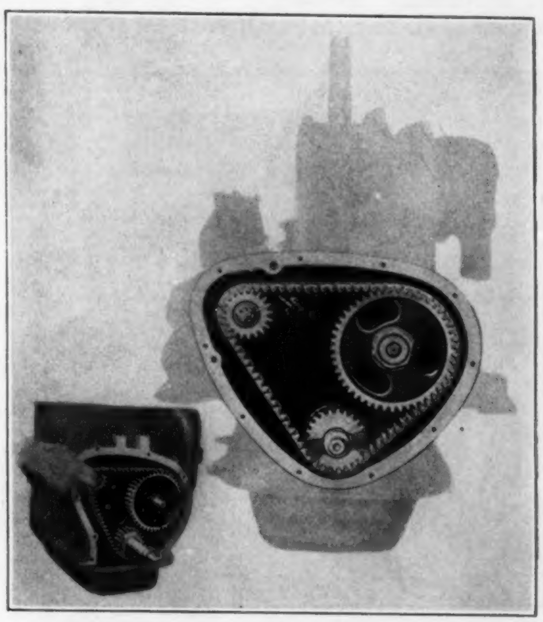


Fig. 8—Special Layout for Camshaft and Generator
The Camshaft and the Generator Are on Opposite Sides of the Engine. The High Position of the Generator Should Be Noted Since It Is Placed There Purposely To Provide Ample Chain-Contact on the Crankshaft Sprocket

sequent period of adjustment is consumed and the chain becomes very slack, it is time to discard it.

The length of life of chain is somewhat a variable, depending upon the nature of the drive and the conditions to which it is subjected. There are drivers who, by abuse and neglect, can drive it to destruction in 10,000 miles; under such conditions, one cannot expect normal service of any of the mechanical parts. Under average conditions, we find that normal chain life is around 30,000 to 50,000 miles.

The type of adjustment to employ depends largely upon the condition of the drive and the amount of money the designer is willing to spend for it. The one shown in Fig. 4 is a manual type conveniently applied to the accessory shaft, and is particularly useful when there are two or more units driven on the same line. It is patented by our company and consists of a main support which has an eccentric extension providing a bearing for the sprocket, and through which the shaft passes central with the mounting. The eccentric relation of the sprocket to the shaft is compensated by means of the grooved floating coupling, engaging both the sprocket and the end flange and passing to and fro at each revolution of the sprocket. The tapered jaws of the couplings are held in perfect engagement by the springs at the rear of the sprocket, bearing against a hardened-steel washer. The chain adjustment is made by rotating the support, thus changing the position of the sprocket with relation to the others and clamping it into position.

The V-8 Drive

In passing to other types of drive, the "automatic-adjustment" type shown in Fig. 5, a design used on a standard high-grade car equipped with the V-8 engine, is interesting. It will be noted that the camshaft only is driven by chain, an extension of which takes the drive-pulley for the fan and generator, with the automatic adjustment used as the third sprocket. The sectional cut of the adjustment, Fig. 5, indicates its method of operation; namely, that the solid support shown, containing the larger coil-springs, forms the common support for the unit, and is attached rigidly to the crankcase. A slide, with a rectangular-shaped hole and forming a retainer for the bushing, fits over the support and is held in place by the three coil-springs. Also, contained in the slide, are two ratchet-jaws engaging grooves of fine pitch that are milled in the support. Thus, a spring tension is always maintained against the sprocket. In operation, the pull of the chain overcomes the tension of the springs and draws the ratchet jaws against the grooves. As sufficient wear takes place in the chain, the ratchet will, of course, engage the next groove while the chain is at rest. It is supplied with oil through the pressure-line.

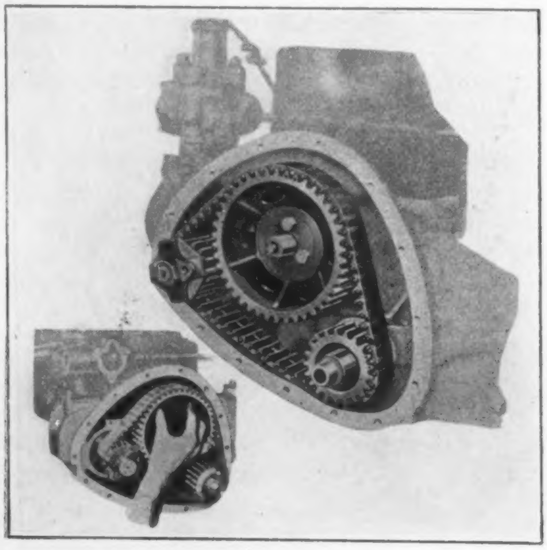


Fig. 9—Assembly with Manual Adjustment
Sometimes It Is Impossible To Assemble the Accessory Unit in Place Last. In This Case It Is Necessary To Flange the Camshaft Sprocket, Attaching It with Cap-Screws over the Short Pilot and Placing It in Position

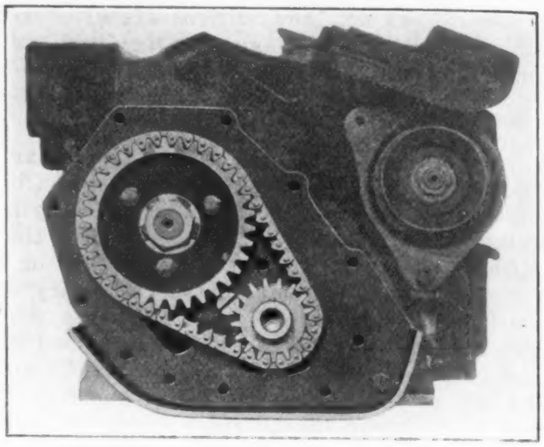


Fig. 10—Two-Sprocket Drive without Adjustment
With the Use of a Chain of Liberal Width, It Is Sometimes Found Possible To Drive without Adjustment Where There Is No Small Driven Sprocket Involved, and the Element of Wear Is Thus Greatly Reduced

Other Types of Chain

Fig. 6 shows the duplex chain. This type of adjustment comes into use very nicely with the duplex chain, or chain which engages the sprocket-teeth on either face. The duplex chain increases the possibilities of chain driving by its advantage in covering shafts of fixed centers or extraordinary designs. Where the accessory shaft is at the right of the camshaft, the chain can be driven over the top side of the crankshaft sprocket, and thence to the accessory sprocket through the automatic idler. The design shown is a $\frac{3}{8}$-in.-pitch chain, $1\frac{1}{2}$ in. wide and 96 links long. The sprockets are 25, 50 and 20 teeth on the crankshaft, camshaft and accessory shaft, respectively, and 33 teeth on the adjustment idler.

The layout in Fig. 7 shows the use of the spring-drive generator-sprocket, in which the sprocket is mounted on a specially designed bracket having an extension that forms the bearing. The slotted, S.A.E. Standard, generator-flange attaches flush to this bracket, which is also slotted, and the chain is manually adjusted by moving the generator outward, pivoting on the lower cap-screw. The generator drive is then taken through a laminated-spring connection that flanges into the end of the sprocket and engages the slotted armature-shaft at the opposite end, the purpose of which is to facilitate removing the generator without disturbing the chain. This, of course, is an advantage in servicing the generator. In addition, the spring drive removes shock load of the generator on the chain, as well as relieving the generator-bearing load common in the

(*Concluded on p.* 160)

Vol. XX January, 1927 No. 1

Brakes for Automotive Use

By John Wiggers[1]

Southern California Section Paper *Illustrated with* Drawings

ABSTRACT

A COMPREHENSIVE review is given of the developments that led to the present status of braking systems for automotive use, including suggestions for minimizing or eliminating the defects in each system, and drawings to illustrate the points covered are presented.

Following an outline of the trend of development, the author describes the hydraulic, the air-pressure and the vacuum types of brake, making special comment on braking systems suited for heavy-duty service. No difficulty has been experienced in stopping the 10-ton six-wheel trucks built by the company the author represents with properly designed mechanical brakes when proper leverages, plenty of brake area and large brake-drums are provided. On these trucks, a service brake comprising four 21-in. brakes having a 5-in. face is used and the brake-bands almost encircle the drum. The foot-brake area alone is thus 1260 sq. in. and, in addition, a propeller-shaft hand brake is provided. A description of the brake linkage used on these trucks is included.

ENGINEERS had not paid much attention to the subject of brakes until a few years ago, and the introduction of four-wheel brakes on motor-vehicles started a remarkable change in the designing of brakes. Then came motorcoaches, which required an entirely different system from that used on trucks and passenger-cars, because motorcoaches travel at relatively high speeds; in city service, they must be able to make quick stops and, in fact, motorcoaches must be able to start and stop as quickly as an ordinary passenger-car. This was not possible with the brakes ordinarily used on trucks. Last came heavy-duty high-speed motor-trucks running on pneumatic tires, carrying heavy loads and pulling trailers. I will treat each of these subjects separately.

Brakes have three general functions to perform: (*a*) to maintain a vehicle at rest, (*b*) to reduce the speed of the vehicle, or bring it to a stop and (*c*) to hold the vehicle to a constant speed while descending a grade. In designing brakes, certain desirable features must be kept in mind, such as the following:

(1) To obtain maximum braking-efficiency with reasonable physical effort on the part of the driver
(2) Smoothness of retardation
(3) A retarding effect proportional to the pedal pressure
(4) Eliminate the tendency to "self lock"
(5) Durability and long life
(6) Ease and simplicity of adjustment
(7) Easy operation and thorough dependability
(8) Brake-operating levers to be within easy reach of the operator

Regarding four-wheel brakes on passenger-cars, Fig. 1 shows the conventional type of front axle before front-wheel brakes were in use. The steering-knuckle is not inclined and there is a considerable distance between the center line of the steering-knuckle pin and the center line of the tire at *a*. It was on an axle of this type that some manufacturers attempted to install front-wheel brakes, but the attempt was not a success. This construction was both unsatisfactory and dangerous because it was impossible to keep the brakes properly equalized and, such being the case, a tendency existed to pull the car to the side of the road. This condition also made steering very hard, and it immediately led to a change in construction of front axle to that shown in Fig. 2, now used on cars having four-wheel brakes.

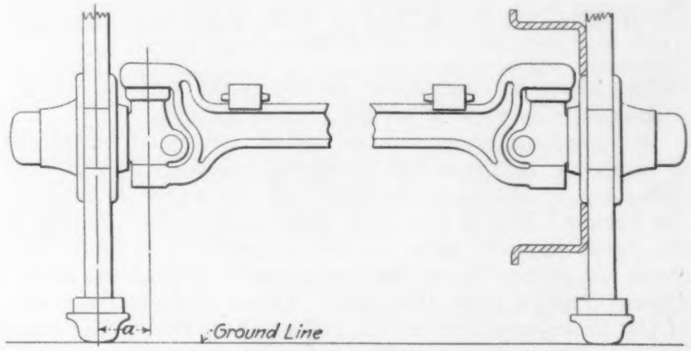


Fig. 1—Former Conventional Type of Front Axle
This Type Was Conventional before Front-Wheel Brakes Were Used. The Steering-Knuckle Was Not Inclined and Is a Considerable Distance from the Center Line of the Steering-Knuckle Pin and the Center Line of the Tire

In Fig. 2 a line is drawn through the steering-knuckle pin and intersects a line through the center of the tire where it touches the ground. Due to the use of this inclined steering-knuckle pin, the braking effect has practically no effect on steering. This design allows practically any amount of steering-lock without causing the car to swerve to one side or the other. After engineers had perfected this part of the brake construction, diffi-

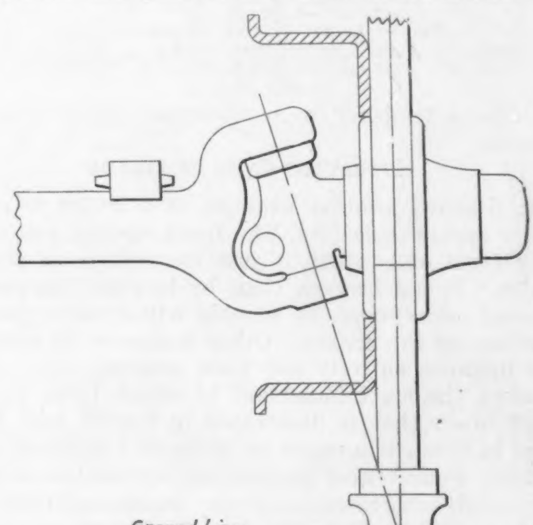


Fig. 2—Front-Axle Construction Now Used on Cars Having Four-Wheel Brakes
Due to the Use of the Inclined Steering-Knuckle Pin, the Braking Effect Has Practically No Effect on Steering

[1] M.S.A.E.—Chief engineer, Moreland Motor Truck Co., Burbank, Cal.

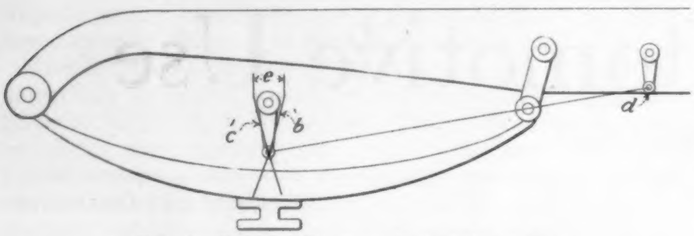


FIG. 3—FORMER TYPE OF BRAKE LINKAGE
The Design Shown Is a Poor One Because the Front Springs Have Too Much Camber and Because the Center Line of the Brake-Rod Is Too High

culties arose in connecting up the brake linkage. Fig. 3 shows a side view of an old type of linkage.

In designing brake linkage, great care must be taken to locate the center of the brake-rods properly so that the deflection of the front springs will not apply or release the brakes. Fig. 3 is a very poor design; first, because the front springs have too much camber; second, because the center line of the brake-rod is located too high. Curves *b* and *c* make this clear. Curve *b* shows the path of the brake-lever due to the deflection of the spring and curve *c*, taken from point *d*, shows that there is a discrepancy between these curves. When the front springs deflect, this distance between curves *b* and *c*, or *e*, is the amount the brakes will be released. In practice, both springs will not always "give" the same amount; hence, a brake-equalizer will have to be installed to equalize both sides.

Fig. 4 shows a better design than that of Fig. 3; first, because the front springs are flat, there being no back movement of axle due to the compression of front springs; second, the brake-shaft is properly located at *f* and, by comparing curves *g* and *h*, it is seen that they almost coincide, hence affecting the movement of the brake-lever very little. Some designers set point *f* higher so that, when the springs do compress, the brakes

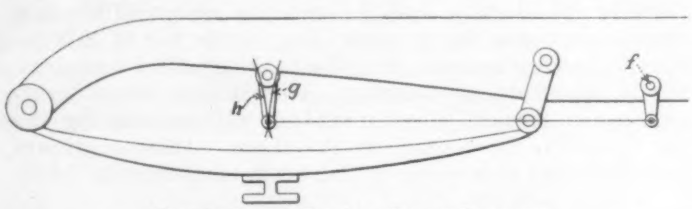


FIG. 4—LATER TYPE OF BRAKE LINKAGE
This Design Is Better Than That of Fig. 3 Because the Front Springs Are Flat and the Brake-Shaft Is Properly Located

will have a tendency to be loosened rather than to be tightened.

HYDRAULIC TYPE OF BRAKE

Fig. 5 shows another example of a layout to compensate for spring deflection. The front springs are shackled on the front end, which is just the reverse of the usual practice. It can be seen that, by locating the center of brake-rod and spring, no trouble will arise in loosening or tightening the brakes. Other designers do away with brake linkages entirely and have adopted other systems of brakes, the most important of which is an hydraulic type of brake that is illustrated in Fig. 6, and was described in detail in a paper by Malcolm Loughead entitled Hydraulic Four-Wheel Brakes for Automotive Vehicles[2]. More briefly described, *i* is the master-cylinder, from which are fed the four actuating brake-cylinders *j*; *k* is the reserve-supply tank, and copper tubing and fittings with flexible connections are used to interconnect the cylinders. The pressure is built up in the master or pressure-cylinder *i*, which is mounted either on the transmission case or on a bracket at the side of the frame. The piston in this cylinder is connected directly to the lower end of the brake-pedal and is forced into the cylinder by pressure on the foot-pedal. The ratio of the movement of the pedal to that of the piston is usually about 4 to 1.

Four copper lines, leading from this master-cylinder to points on the frame opposite the brake-actuating cylinders *j*, are located in each wheel. The connection between these cylinders and the end of the copper tubing is a suitable piece of flexible hose. This hose is reinforced on the inside by a close-wound coil-spring. This is necessary because no expansion must take place in this hose. The flexible connection is necessary to take care of two movements, those due to steering and those due to the spring action. An enlarged view of the master-cylinder is shown in Fig. 7. Great care is necessary in the selection of the liquid used in this system.

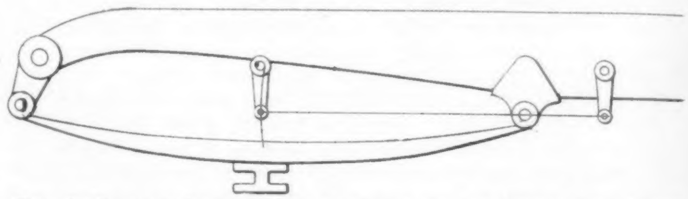


FIG. 5—LAYOUT DESIGNED TO COMPENSATE FOR SPRING DEFLECTION
The Front Springs Are Shackled on the Front End, Which Is Just the Reverse of the Usual Practice. By Locating the Center of Brake-Rod and Spring, No Trouble Will Arise in Loosening or Tightening the Brakes

The liquid used consists of a mixture of No. 1 castor-oil and formula No. 1 denatured alcohol. This mixture usually consists of one part of alcohol to two parts of castor-oil, but the proportion varies in different parts of the Country.

The company that manufactures this braking system states that some users have been having trouble because the fluid in the supply-tank became congealed. This thickening of the fluid gives it the appearance of a gray soapy mass; it is caused by the action of the castor-oil on the plating in the supply-tank. Tests have shown that adding 0.75 grams of sodium arsenite to 1 gal. of brake-system fluid, effectively prevents the formation of rust in this system. Any leakages are taken care of by a small reservoir, *k*, mounted on the dash; by opening a valve in the bottom of this tank, any leakages can be replaced. The only trouble I have experienced with this

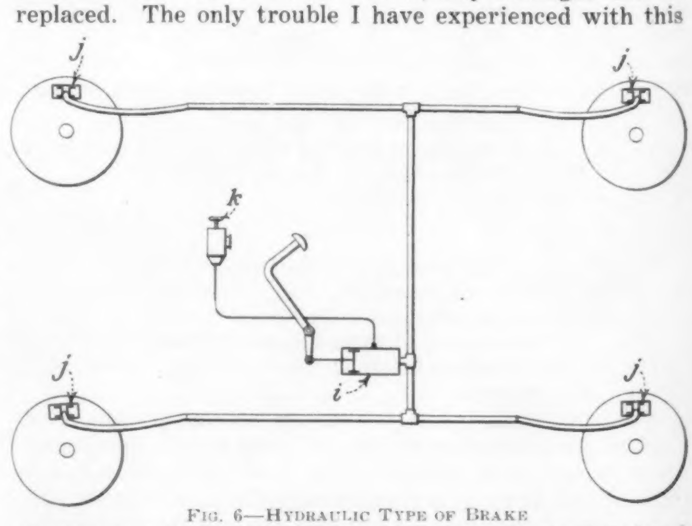


FIG. 6—HYDRAULIC TYPE OF BRAKE
The Master-Cylinder *i* Feeds the Four Actuating Brake-Cylinders *j*, and *k* Is the Reserve Supply-Tank. Copper Tubing and Fittings with Flexible Connections Are Used To Interconnect the Cylinders

[2] See THE JOURNAL, October, 1923, p. 313.

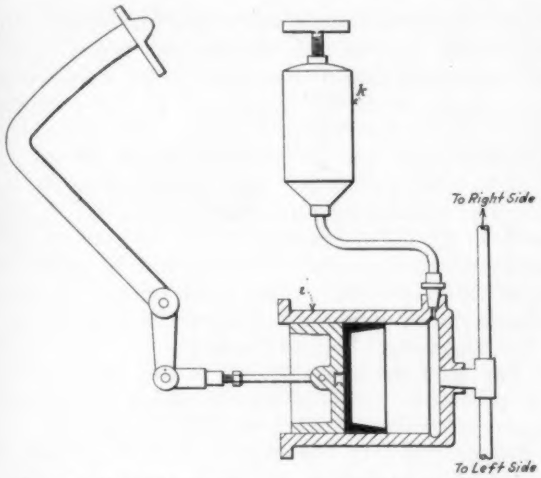


FIG. 7—MASTER-CYLINDER OF THE HYDRAULIC TYPE OF BRAKE

The Fluid Used in the System To Transmit the Pressure Usually Consists of a Mixture Composed of One Part of Alcohol to Two Parts of Castor-Oil

system on passenger-cars, and on some trucks, was due to leaky cup-leathers, but as these are now made from special molded-rubber material, very little trouble is experienced with the ordinary pressure employed. The normal working pressure of this system is 250 lb. per sq. in., although it is physically possible to exert a pressure of 500 lb. per sq. in. in case of emergency.

Another type of hydraulic brake recently developed is known as the low-pressure hydraulic system, the average working pressure being around 25 lb. per sq. in. Generally speaking, this system is similar to the type shown in Fig. 6, except that it has expanding brake-shoes. The system is shown in Fig. 8 and operates as follows: A

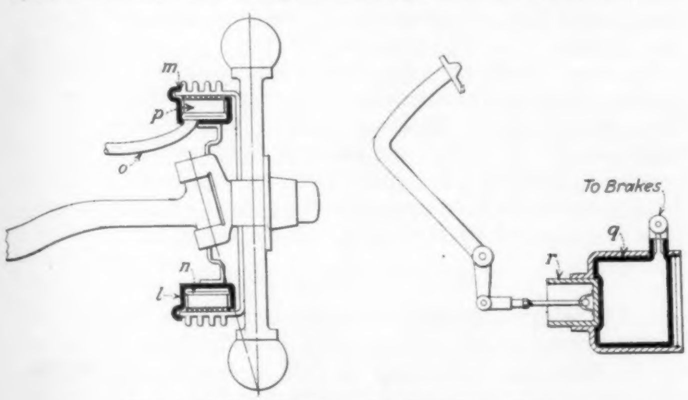


FIG. 8—ANOTHER TYPE OF HYDRAULIC BRAKE

In This System, Water Is the Fluid Used To Transmit the Pressure. It Is Similar to the Type Shown in Fig. 6, Except That It Has Expanding Brake-Shoes

channel-shaped brake-shoe holder, *l*, of pressed steel is fastened to the steering-knuckle and, to make an enclosure to keep the water and dirt out, the inner flange is turned around the inner-edge of the brake-drum shown at *m*. A heavy-walled rubber-tube, *n*, resembling a small-diameter inner-tube, is carried in the bottom of this channel, and from it extends an integrally molded length of rubber tubing, *o*, properly reinforced to keep it from expanding. This tube connects with the copper-tube lines on the frame, as in the system illustrated by Fig. 6. On the outside of this rubber tube are six sectional-shoes *p* of pressed steel, upon which the brake-lining is riveted. Between each shoe is a pin to keep the shoes from turning and a spring to hold them against the rubber tube. The master-cylinder is operated by the foot-pedal, the same as in the system shown in Fig. 6, the only difference being that the master-cylinder is replaced by a large rubber-bag, *q*, and a piston, *r*, about 3 in. in diameter, which forces the liquid from this bag to the rubber tubes mounted in the wheels. The liquid used in this system is water and, in cold climates, an anti-freezing solution must be employed. So far as I can determine, this brake is used at present only in the Stutz Vertical-Eight, although I know it is being experimented with by other companies. The advantages claimed for this system are, first, that it has a low working-fluid pressure of 25 lb. per sq. in. as compared with the usual 250 lb. per sq. in. in the system shown in Fig. 6. This low pressure is sufficient because the pressure is equalized on all parts of the inner tube so that each shoe is forced out with even pressure. A second advantage claimed is that it is a self-contained system, there being no pistons or cup leathers to leak, and a third claim is that, being a low-pressure system, the foot-pedal pressure is lower.

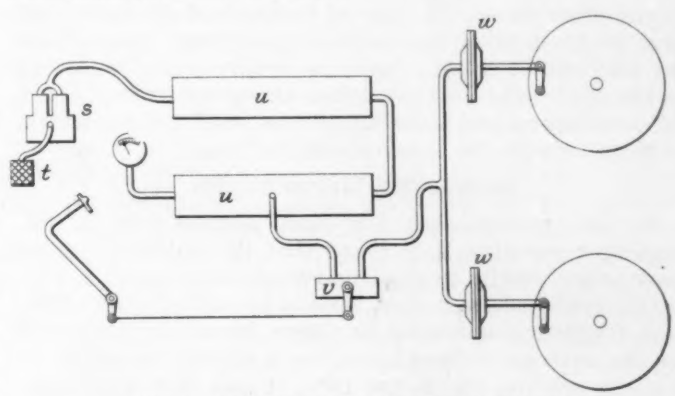


FIG. 9—WESTINGHOUSE TYPE OF AIRBRAKE

The System Usually Consists of an Air-Compressor, *s*, with an Air-Cleaner, *t*, Two Storage-Tanks *u* Fitted with a Pressure-Relief Valve and Control-Valve, *v*, and Two Brake-Chambers *w*, Operating Diaphragms, and Necessary Pipes and Fittings

A number of cars use mechanical brakes and are employing brakes of multiple-shoe type or of the servo-brake type to get an easy pedal-operation. In this type the pressure is applied to the first shoe, which, through the action of the revolving wheel, transmits the pressure

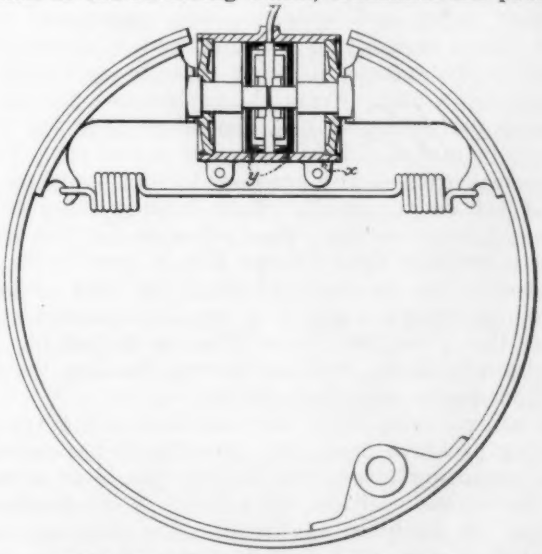


FIG. 10—CHRISTENSEN TYPE OF BRAKE

Its Operation Is Practically the Same as That of the Westinghouse System Except That, in Place of the Diaphragms, Brake-Cylinders Are Fastened Directly to the Brake-Bands. Fastened to the Brake-Anchor Is a Cylinder, *x*, Having Two Pistons *y*. These Pistons Push Directly against the Ends of the Brake-Bands. The Pressure Is Admitted at a Point between the Two Pistons

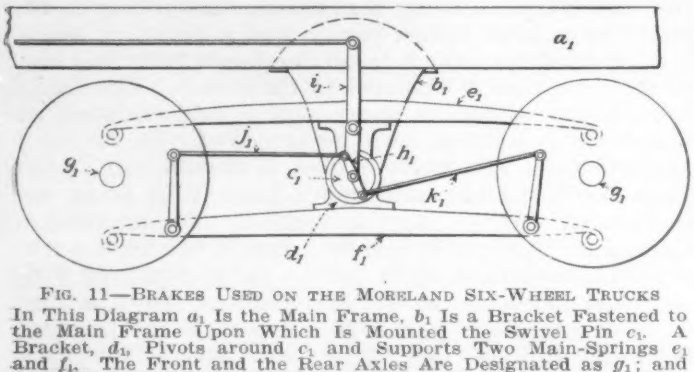


FIG. 11—BRAKES USED ON THE MORELAND SIX-WHEEL TRUCKS

In This Diagram a_1 Is the Main Frame, b_1 Is a Bracket Fastened to the Main Frame Upon Which Is Mounted the Swivel Pin c_1. A Bracket, d_1, Pivots around c_1 and Supports Two Main-Springs e_1 and f_1. The Front and the Rear Axles Are Designated as g_1; and h_1 Is the Equalizing-Bar Which Is Fastened to the Lever i_1 That Is Attached to the Brake-Rod Coming from the Foot Brake-Pedal. The Center Line of the Lever h_1 Is Directly over the Center Line of Pin c_1. This Is Necessary To Take Care of the Circular Motion of Axles and Springs around Pivot c_1. The Front and the Rear Axles Can Swing Up or Down without Affecting the Brake-Operating Rods j_1 and k_1

to the other shoes. A type of brake used on passenger-cars by Studebaker has a small gear-pump mounted on the rear end of the transmission, with a valve connected to the foot-pedal so that, when the pedal is depressed, the valve opens and forces oil to two small pistons which, in turn, actuate the brake operating-lever.

BRAKES FOR MOTORCOACHES

On all motorcoaches for rural service, up to 30-passenger capacity, it is found that the ordinary brakes work satisfactorily to stop the vehicle within safe limits. In city traffic where brakes have to be applied very often, such frequent application has been found too fatiguing for the average driver; hence, on some motorcoaches of this size and on the larger type, it has been found advisable to install some type of power-brake. The most widely known of these is the Westinghouse type of airbrake shown in Fig. 9. It is described in detail in a paper by H. D. Hukill entitled The Automotive Airbrake—Why and How[3], and its operation is described briefly as follows: This system usually consists of an air-compressor, *s*, with an air-cleaner, *t*, two storage-tanks *u* fitted with a pressure-relief valve and control-valve, *v*, and two brake-chambers *w*, operating diaphragms, and necessary pipes and fittings. The compressor is air-cooled, has a capacity of 3 cu. ft. and is driven by the engine of the vehicle, it being directly connected by a chain or by a belt. From the compressor the air goes to one or two storage-tanks, depending upon the volume of air to be stored. These tanks are made from seamless drawn-steel, welded, and are tested to an hydraulic pressure of 600 lb. per sq. in. Each tank is fitted with an ordinary type of safety valve, adjusted to blow off at a pressure ranging from 150 to 200 lb. per sq. in. The control-valve can be operated either by hand or by foot but, in motorcoach work, it is usually operated by foot because this gives the driver freedom to use his hands for collecting fares, making change, issuing transfers, operating doors, and other duties.

The control-valve is of the balanced-spring type, the operating pressure depending directly on the movement of the operating lever; the farther the lever is moved from the release position, the greater is the pressure on the line. In applying the brakes, the pressure can be graduated in steps of 1 or 2 lb. per sq. in. The success of airbrake equipment depends upon the flexible and sensitive graduated-control valve. The maximum brake-pressure in the line depends upon the stiffness of a regulating-spring; pressure is usually around 50 to 75 lb. per sq. in. The brake-chambers *w* are not of the cylinder or piston type, but of the diaphragm type; each diaphragm consists of two dish-shaped plates between which the diaphragm, made of a special rubber-composition with fabric-layer inserts, is located. The pressure-line is admitted to one side of the diaphragm; on the other side, a plate and a push-rod are connected to the brake rigging. The diaphragms are made in various diameters of from 5 to 11 in., depending upon the size of vehicle. The maximum pressure at the push-rod is from 300 to 1500 lb. at 50-lb. per sq. in. pressure on the diaphragm. The advantages of using diaphragms instead of pistons is that the brake-chambers are entirely free from leakage without any packing and respond to the slightest variations in pressure. The brake-chambers *w* require no lubrication or maintenance throughout the life of the rubber diaphragm. In the Westinghouse system, these diaphragms usually are mounted on the axle or on the frame.

Another type of brake system very similar to the Westinghouse system is known as the Christensen type. Its operation is practically the same as that of the Westinghouse system except that, in place of the diaphragms, brake-cylinders are fastened directly to the brake-bands. This construction is shown in Fig. 10. Fastened to the brake-anchor is a cylinder, *x*, having two pistons *y*. These pistons push directly against the ends of the brake-bands. The pressure is admitted at a point between the two pistons. This system does away with all brake-levers, rods, cams, and the like and is claimed to be more efficient because of the absence of any friction in the brake linkage. Its disadvantage may be that it has cup leathers which are liable to leak. On its six-wheel motorcoaches, the company I represent uses a combination; that is, the pressure-valve and the storage-tanks are of the Westinghouse type and the brake-operating system is of the Christensen type.

On heavy-duty trucks, mechanical brakes or power-brakes, such as the airbrake, the vacuum-brake and the like, may be used. The most widely known is the Westinghouse system, the same as used for heavy-duty motorcoaches. However, for trailer service, the trailer is fitted with a reservoir tank and has an application release-valve that is operated by the main control-valve.

VACUUM TYPE OF BRAKE

There are two makes of vacuum-brakes now in use, one is known as the B-K system, and the other is known as the Hercules system. The B-K system has a vacuum-cylinder, mounted at a convenient location on the frame, connected directly to the brake leverage. This cylinder is operated by vacuum induced by the intake-manifold;

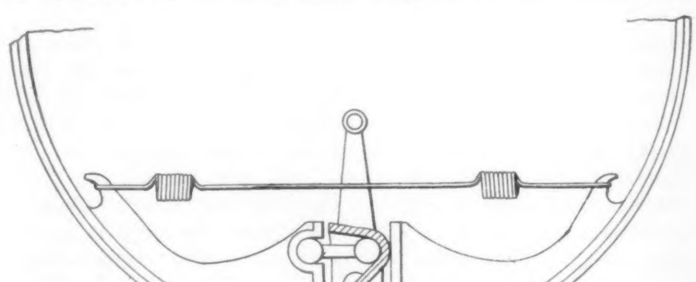


FIG. 12—ENLARGED VIEW OF SIX-WHEEL-TRUCK BRAKE-CAM

The Small Links Shown at l_1 Are only 9/16 In. from the Center Line of the Brake Camshaft and the Leverage Effect Is, Therefore, Very Great

[3] See THE JOURNAL, March, 1925, p. 283.

connection from the manifold and one end of the tank is controlled by a valve that is operated by the foot-pedal. The farther the foot-pedal is depressed, the greater is the vacuum that operates the brakes.

In the Hercules system there is a large tank from which the air is withdrawn by the intake-manifold in a manner similar to that of the B-K system. When the brakes are applied, the air is drawn from the operating cylinder into the large tank, the idea being that this vacuum-tank is large enough to take care of a great number of brake applications, and it actuates the operating cylinder more rapidly than if direct connection were established between the intake-manifold and the operating-cylinder. For trailer service a small tank and an operating-cylinder are installed on the trailer, all being operated by one control-valve on the truck.

The company I represent has solved the problem of brakes on its 10-ton six-wheel trucks. It has found that, by providing proper leverages, plenty of brake area and large drums, no difficulty is experienced in stopping its largest truck with a properly designed mechanical brake. The ordinary heavy-duty four-wheel truck, equipped with Timken duplex-brakes, has a braking-surface of approximately 300 sq. in., while, on the 10-ton six-wheel truck used by our company, a foot-brake of four 21-in. brakes with a 5-in. face is used and the brake-bands almost completely encircle the drums. The foot-brake friction area alone is thus 1260 sq. in. and, in addition, we provide a propeller-shaft hand-brake.

Six-Wheel Truck Brake-Linkage

It is a problem to keep the brake linkage arranged at all times so as not to interfere with the brake adjustment. Fig. 11 shows a sketch of the rear end of our six-wheel truck; a_1 is the main frame, b_1 is a bracket fastened to the main frame upon which is mounted the swivel pin c_1. A bracket, d_1, pivots around c_1 and supports two main-springs e_1 and f_1. The front and the rear axles are designated as g_1; and h_1 is the equalizing-bar which is fastened to the lever i_1 which is attached to the brake-rod coming from the foot brake-pedal. The center line of the lever h_1 is directly over the center line of pin c_1. This is necessary to take care of the circular motion of axles and springs around pivot c_1. The front and the rear axles can swing up or down without affecting the brake-operating rods j_1 and k_1.

Another condition we had to contend with in this design was that, as the springs flatten, they naturally become longer; but the equalizing-bar h_1 compensates for this as since both halves of the springs lengthen, the lever automatically takes care of this by swinging around its center.

We make no attempt to equalize the brakes on the right and the left side of the chassis. We find that, by properly adjusting each of the two sides, they can be made to pull equally. We think it is better practice not to equalize from one side to the other because if the full-equalizing system is used and a single pin should become disconnected, the entire braking system becomes inoperative. We have a number of these trucks running over steep mountain-ridges and the brakes are of such dimensions that one side will stop the car even though the other side should be out of commission. For this reason we have eliminated "full equalization." Fig. 12 is an enlarged view of the brake-cam. The small links shown at l_1 are only 9/16 in. from the center line of the brake camshaft and the leverage effect is, therefore, very great.

THE DISCUSSION

Eugene Power[4]:—Regarding brake-drum area, is it correct to say that the greater the diameter of the drum is, the less is the effort required on the pedal? If the width of the drum is increased, is there any limit to such increase to balance the fact that the diameter of the brake-drum may be limited?

Ethelbert Favary[5]:—With a wider brake-drum, a larger area of brake-lining is available and the pressure per square inch is less; hence, the brake-linings last longer, but width of brake-drum does not affect the pressure to be applied to the brake-pedal since it does not change the total frictional-force required, other things being equal and considering the linkage to be the same in one case as in the other. For example, if a 100-lb. weight, rests on an area of 100 sq. in. of brake-lining, the pressure is 1 lb. per sq. in. If the coefficient of friction is 0.3, a 30-lb. pull is needed to drag the weight along. If the same weight rests on an area of 10 sq. in. instead of on an area of 100 sq. in., it still requires a 30-lb. pull to drag it along that surface, because the downward pressure is now 10 lb. per sq. in. and the area multiplied by the pressure is still 100. If the area of bearing-surface is too small, the lining will heat and the coefficient of friction will drop if the temperature rises beyond a certain point. This is true of all brake-linings.

This raises the interesting question whether a narrow tire will skid more than a wide tire. In 1900, the London Electric Cab Co. used solid tires $1\frac{3}{4}$ in. wide and these wore out after running about 2000 miles. When the width was increased to $2\frac{1}{2}$ or 3 in. the drivers complained that the cabs skidded from one side of the street to the other. I stated that the total frictional-force remains the same if the conditions remain the same; but, in London there is a film of mud on the pavement during 8 months of the year, due to fog and rain. When the tire was $1\frac{3}{4}$ in. wide, it dug right through that film of mud and gripped the pavement itself. When the tire width was increased, it evidently did not squeeze out all the mud from between it and the pavement and the coefficient of friction was considerably smaller.

With balloon and with pneumatic tires we find that the pressure per unit area is very much reduced and yet, the wider they are the less they skid on certain roads. A balloon tire skids less than an ordinary pneumatic tire; first, because it has a greater number of non-skid projections, the edges of which grip the ground, and the pressure per square inch at the contact with the road is so high that it squeezes all the mud out and reaches the dry surface. A second reason is that roads are composed of ripples, and probably a larger portion of a balloon tire accommodates itself to those ripples than is true of a pneumatic tire.

Mr. Power:—What are the relative merits of single solid and dual solid-rubber tires as to prevention of skidding?

Mr. Favary:—By using the dual tires there are two corners or edges to grip the road surface and perhaps that will retard skidding.

Mr. Power:—I contended that we should use dual tires, but one of our men said he obtained better results with single tires.

Mr. Favary:—Instead of having a single tire of certain width, it is possible to use dual tires and keep the total width of tire surface the same. The dual tires should be softer than the others because there is more

[4] M.S.A.E.—Superintendent of automotive equipment, Union Oil Co. of California, Los Angeles.

[5] M.S.A.E.—Consulting engineer and manager of sales promotion, Moreland Motor Truck Co., Los Angeles.

room for the rubber to spread; that is, a flat piece of rubber will "give" less under a certain load than if that piece of rubber is divided so that the rubber has room in which to spread.

MR. POWER:—If it starts to slip, a wide solid-tire takes the mud along with it. If a dual tire slips, one of the dual tires has cleaned the street surface before the other dual tire has slipped onto that portion.

E. B. MOORE[6]:—We used 12-in. single-tires but had to take them off and put on dual tires in wet weather. The non-skid single-tires were not as good as plain-tread dual-tires in actual practice.

A MEMBER:—Dual tires stop the side skids. With balloon tires, I think there is no difference between the wide tread and the narrow tread on a clean street, wet or dry, but on a street that is dirty and dry, balloon tires skid much more than the high-pressure tires. A balloon tire skids on a piece of paper and it is more difficult to adjust the brakes for balloon tires and especially so for semi-balloon tires.

MR. FAVARY:—It depends largely on the road surface as to whether a balloon tire skids more or skids less. In tests made on certain roads, balloon tires acted better and, on other very smooth and hard roads, they did not act well because the pressure was less and they did not squeeze out all the mud that was between them and the road surface.

A MEMBER:—Balloon tires sway on a dry street, side skid and skid forward.

MR. FAVARY:—A balloon tire skids considerably more forward than it does sidewise for the treads scrape the pavement clean. Numerous tests show certain results and numerous other tests show different results, largely due to the type of anti-skid tread, the road, weather conditions and the composition of the rubber.

I have heard from some manufacturers of brake-linings that, according to their experience, if the coefficient of friction is too great, it is likely to cause the band to grip or lock when that is not wanted. All manufacturers say they can give brake-lining any coefficient of friction desired, depending upon the proportion of asbestos put in, but that they do not wish the coefficient of friction to be too great, for the reason stated.

In tests that I have witnessed, the coefficient of friction was low at first and rapidly rose. There happened to be considerable moisture in the atmosphere, and I believe that the low coefficient was due to moisture on the drum because, as soon as the moisture was dried up, the coefficient of friction rose.

A. N. DAY[7]:—Regarding the skidding of dual tires as compared with that of solid tires, it has been our experience as tire builders that skidding is dependent upon the ability of the tire to conform to the irregularities of the road. Comparing dual tires with one-piece solid-tires, we have found that the dual tires give better tracks on wet pavements than the one-piece solid-tires and I believe Mr. Moore's experiments bear that out.

MR. FAVARY:—Skidding on a steep hill can be lessened by letting air out of the tires and reducing the inflation pressure.

MR. DAY:—During the World War, ambulances used large 6, 7 and 8-in. tires and, when they stalled, the drivers let some of the air out of the tires, rolled out and onto harder ground, reinflated the tires and went ahead.

QUESTION:—Do the devices used for attaching external brake-bands and the devices for actuating them maintain uniform pressure all around the band, or is the pressure much heavier on one side than on another?

MR. FAVARY:—Some engineers claim that they do maintain uniform pressure but it is doubtful whether that is actually true in practice.

PROPELLER-SHAFT BRAKES

A MEMBER:—What are the comparable efficiencies of the propeller-shaft brake and the brakes on the wheels of the 10-ton six-wheel trucks you are building? I believe there are double brakes on all four rear-wheels.

JOHN WIGGERS:—The principal object in using a propeller-shaft brake is to use the drums in the wheels so as to get the maximum diameter of drum and width of braking-surface connected to the foot-brake. We have attempted to make the foot-brake do most of the work, and to use the hand-brake for holding and for emergency needs. We have used the rear wheel-drums entirely for the foot-brake. Regarding the propeller-shaft brake, there being a reduction between the rear wheels and the propeller-shaft, a propeller-shaft brake of smaller diameter in proportion to that of the rear brake-drums can be used and still get the same efficiency.

MR. FAVARY:—Supplementing what Mr. Wiggers has just said, my experience has shown me that for heavy loads the transmission or propeller-shaft brake should not be used as a service or foot-brake because, when the propeller-shaft brake is used, great stresses are transmitted through the entire driving-mechanism. With heavy loads, it is not possible to skid the tires when the clutch is thrown in whereas, with light cars, that can be done readily. For that reason, the stresses in heavy trucks in connection with heavy loads are, as a rule, greater in the driving mechanism than the stresses coming from the engine. When very powerful engines are used which have the power to slide the rear wheels under full loads, then the stresses are about the same in one case as another; but, ordinarily, the propeller-shaft brake, on account of its ratio, is more powerful. When the foot-brakes are used, the stresses are applied to the wheels themselves and do not go through the driving-mechanism or the rear axles; hence, I think it is preferable with heavy loads to use the transmission brake only for the emergency or hand-brake.

COOLING AND OTHER EFFECTS

QUESTION:—What considerations govern the size of brake-bands for dissipation of heat, and what temperatures are permissible in various types of installation?

MR. WIGGERS:—Ordinarily, an engineer tries to design the brake-drum so as to have it of as large diameter as possible. This is governed to a considerable extent by the size of tire. Practically, up to this year, the majority of tires were 34 x 5, 36 x 6 and 38 x 7 in. All the above sizes had what is known as a 24-in. base rim; whereas, this year, the sizes have dropped to 30 x 5, 32 x 6 and 34 x 7 in., which gives the tires a base of 20 in. Therefore, it is necessary to make the brake-drum of smaller size than heretofore.

QUESTION:—How effective are radiating ribs in cooling the brake-drums?

MR. WIGGERS:—They act as a cooling medium and help to keep the drum in shape, aiding the cooling effect considerably.

R. H. MCNEISH[8]:—My experience has been that balloon tires give more braking-effect, especially in wet weather, due possibly to a vacuum created by the non-skid tread. In dry weather I believe they hug the road closer, on

[6] M.S.A.E.—Superintendent and manager, Los Angeles Automotive Works, Los Angeles.

[7] Sales engineer, Overman Cushion Tire Co., Los Angeles.

[8] M.S.A.E.—Automotive engineer, Fire Department, City of Los Angeles.

account of the low pressure, and create a higher efficiency in friction. The many small projections, I believe, tend to clean the road surface of foreign substances.

MR. FAVARY:—When traveling over a rough road on hard tires, they are up in the air much of the time. The braking efficiency is reduced and it takes a longer distance to stop. A softer tire hugs the road better. On smooth roads, a hard tire probably slides farther than a balloon tire; so the road surface and the composition of the road have considerable to do with it.

A MEMBER:—When a front-wheel-drive machine is braked, why does it not skid in cases where a car with rear-wheel drive does skid?

MR. FAVARY:—Front-wheel brakes do not permit a car to skid as readily as do rear-wheel brakes. The center of gravity in any vehicle is always some distance from the ground and, if the vehicle tends to stop, there is a tendency for the load to tilt forward. Applying a brake to the front wheels will stop a car much more readily, and it will not skid so easily as when the braking is done through the rear wheels. Braking through the four wheels gives the best results. I cannot see why the front-wheel-drive car should skid less than one with rear-wheel drive unless the weight on the front wheels is greater than that on the rear wheels. It is all a matter of distribution of load on the driving-wheels and the braking-wheels. Efficiency of traction depends on the amount of load on the traction wheels, and efficiency of braking depends on the proportion of load on the braking-wheels.

CLIFTON R. ROCH[9]:—A front-wheel-drive car of mine weighs 2600 lb.; 1400 lb. in front and 1200 lb. in the rear. I can use brakes on either the front or the rear wheels. If the rear wheels slide, the car does not turn around as a car does that is driven by the rear wheels. A small percentage more weight is in front than is in the rear and, when the car slides, it slides straight ahead. One can see that, by putting a transmission brake on my car, I would have nearly a perfect four-wheel brake.

[9] M.S.A.E.—Consulting automotive engineer, Los Angeles.

[10] Instructor, southern branch of the University of California, Los Angeles.

CHARLES PAXTON[10]:—When stopping after having been traveling very rapidly, a cloud of smoke sometimes comes from the rear tires of a car. It seems that, in such a case, the coefficient of friction must be reduced. Have any experiments been made as to whether that effect is due more to the road or to the tire? Under apparently the same condition one tire might heat up and burn, while the other tire would cut.

MR. FAVARY:—Tests have shown that the speed of the car does not affect the deceleration, and that tends to show that the coefficient of friction remains the same. I do not know whether any experiments have been made with rubber that was heated on the road, to determine if its coefficient of friction decreases. It may remain the same. Why the tire on one wheel burns more than the other does may be due to the fact that one side of the street is rougher than the other, or one tire may be composed of a different grade of rubber, or one brake may be more effective than the other. Perhaps one wheel stops and the other rotates.

MR. DAY:—With some brake-linings, when the brakes are applied in the usual manner the car continues to move forward, but with the same brake adjustment and with a different lining, the car stops. What is the explanation of the fact that the former effect is caused by a hard and the latter effect by a soft brake-lining?

MR. FAVARY:—The difference is due to the difference in the coefficient of friction of the brake-lining. One has a higher coefficient than the other; therefore, the same application of power will cause a greater retarding effort at the wheel of one than the other.

A MEMBER:—A soft brake-lining is inclined to give a softer stop with less pedal-pressure when new, but a 1/4-in. lining applied to the band will compress to 3/16 in. after 2 months of hard service. The impregnating compound may dry out of the lining, causing it to glaze. Linings used on several English cars are practically as hard as steel, and still they produce a soft easy stop; so, with an ordinary lining on the brake-band, if the brake does not stop the car it usually is due to glazed brake-lining. After the impregnating compound dries out, nothing is left but raw asbestos.

THE AGRICULTURAL SITUATION

THE crops now are practically accounted for, and in volume leave little to be desired. The total acreage in cultivation was about 2.0 per cent larger than that harvested in 1925, and the combined yield, as calculated by the Department of Agriculture, is 3.4 above that of last year and 3.0 per cent above the average of the last 10 years. The low prices that have been prevailing for farm products and the reported trend of population from the farm to the cities apparently have not as yet had the effect of reducing production in the aggregate. An especially noteworthy feature in view of the somewhat sensational reports that were in circulation a few years ago about the migration of negroes from the rural districts of the South is that this region should now produce a cotton crop more than 2,000,000 bales larger than it ever produced before.

Although, as in many other years, results are more or less uneven over the Country, the trend is in the way of improvement, and on the whole the outcome must be considered good. Since the increase of population in this Country is at the rate of less than 1.5 per cent per year, that the rise in prices in farm products has not been more pronounced, in view of the fact that production has been increasing, is not inexplicable. That the war gave a great stimulus to agricultural production in all countries outside of Europe is well known, and neither in the competing countries nor in the United States has any considerable curtailment of production taken place since the war. If the farms have been losing population, the remaining farm population must be gaining in productive efficiency or the weather conditions have been more favorable. Either explanation would imply lower costs of production, while values on the whole are rising.

While the trend of aggregate value is not far from what might be expected in view of the relation of production to population, the prices of individual products have fluctuated continually in close response to the relations between supply and demand. Wheat, corn, oats, cotton, sugar, potatoes, wool, live stock, and dairy products have moved separately and independently, apparently governed in each case by the master influence. If on the whole production is so large as to make prices unduly low, the necessary conclusion would seem to be that more people than are needed to maintain the desired price situation are still on the farms.—National City Bank of New York.

Headlighting Symposium at 1926 Semi-Annual Meeting

THE discussion of automobile headlighting at the Semi-Annual Meeting, at French Lick Springs, Ind., last June, again took the form of a symposium, similar to those which had previously taken place at the 1926 Annual Meeting[1], and at a meeting of the Metropolitan Section[2]. Other papers on this subject have been submitted by H. M. Crane[3] and A. W. Devine[4]. In the present symposium, papers were presented by R. N. Falge, J. H. Hunt, Mr. Devine, Mr. Crane and Dr. C. H. Sharp. Other discussion followed.

HEADLIGHTING REQUIREMENTS AND AVAILABLE TESTING-APPARATUS

BY R. N. FALGE[5]

ABSTRACT

THE apparatus designed and made available for reaching a conclusion as to what constitutes desirable practice in light intensity and distribution is described. This was so constructed that beams of different depths could be built up and the relative intensities from top to bottom could be varied as desired to spread out the light through various total angles differing from top to bottom and having varying proportions near the center and at the wider angles. The cars supplied by the committee included four head-lamps that were provided with standard variable reflectors, incandescent lamps and plain glass covers in the door. Focusing-screws modified the beam by moving the light-source up and down and along the axis of the reflector. The beams could be spread laterally from 6 to 40 deg. and vertically from 2 to 4 deg. Focusing adjustments allowed deepening of the beams from the top to the bottom. Masks of several kinds increased the depth of the beam and reduced the light output from the head-lamp without changing the color. The results of variations of settings were studied with the car in motion at different speeds when driven over roads of different kinds and under various conditions.

ONE part of the program on which the Joint Steering Committee is working is a symposium of opinion on ideal headlighting. A subcommittee was appointed to design and make available facilities that would make it possible for engineers of motor-car companies to reach a conclusion as to what constitutes desirable practice in light intensity and distribution when the interests of all users of the highway are taken into account and to demonstrate this conclusion. The conclusion reached in comparing the beams of a limited number of commercial equipments is usually not what the observer would choose, if he were given an opportunity to modify the road lighting at will over a wide range and under a great variety of conditions. Fortunately, the equipment necessary to produce these modifications is neither extensive nor complicated and can be quickly attached to or removed from a car.

Headlighting requirements are determined by a large number of factors, each varying greatly from the others. They differ for level and for hilly roads, for straight and for winding highways, for road surfaces having different color, smoothness and reflection characteristics, and for various conditions of road boundary. They depend upon traffic, the illumination from other cars and the glare from approaching vehicles, upon atmospheric conditions in marked degree, speed, spring deflection with loading and car springing, and upon the vision of the driver, his alertness and response. Obviously, a determination of the headlighting that is best for meeting all these conditions cannot be made in one evening or on roads of restricted area. But a short experience with variable equipment soon suggests the range of beam characteristics to be investigated in meeting the demands of typical and difficult situations.

DISTRIBUTION OF LIGHT INTENSITY

The equipment should make it possible to build up beams of different depths and to vary the relative intensities as desired from top to bottom, to spread out the light through various total angles differing from top to bottom and with variable proportions near the center and at the wider angles. One should be able to distribute the light differently to the right and to the left of the car. Alternative lighting should be available when meeting other cars.

Seven cars available for demonstration have equipment supplied by the committee. Included are four head-lamps, provided with standard parabolic reflectors, incandescent lamps and a plain cover-glass in the door. The test lenses and masks slip into clips on the outside of the door to facilitate changes. Two focusing-screws are supplied, one to modify the beam by moving the light source up and down, the other by moving it along the axis of the reflector. The head-lamps can be aimed individually or as a group. A panel is provided with switches, ammeter connections and rheostats for varying the light output from the individual units while driving. This part of the equipment may, for sake of convenience, be built into the instrument-board.

For lateral distribution of the light, 22 lenses are available, with spreads varying in small steps from 6 to 40 deg. For vertical distribution, several means can be employed. For simple vertical spreading, lenses of 2, 3 or 4 deg. are provided. There are also some diffusing covers. The focusing adjustments allow deepening of the beams from the top to the bottom. Six masks of

[1] See THE JOURNAL, July 1926, p. 100.
[2] See THE JOURNAL, December 1925, p. 559.
[3] See THE JOURNAL, May 1926, p. 467.
[4] See THE JOURNAL, July 1926, p. 29.
[5] M.S.A.E.—Engineering department, in charge of automotive lighting, National Lamp Works of the General Electric Co., Nela Park, Cleveland.

several forms are available for decreasing the depth of the beam and reducing the light output from the head-lamp without changing the color.

AUXILIARY DRIVING-LIGHT

To give complete freedom in forming the desired main driving-beam, an auxiliary driving-lamp is included to provide road illumination for passing that will be of a character acceptable to an approaching driver. The top of the beam from this unit should be aimed to intersect the road about 75 ft. ahead of the car.

No fixed procedure is suggested for studying the headlighting problem with these facilities, which are adaptable to the ideas of the different investigators. One way of beginning the work is to drive to a spot where the road conditions are of a certain type and, with the car standing, to build up by trial a distribution that would seem to meet the situation best. If the beam is made up of several overlapping vertical sections, narrow and deeper ones, with lenses of different spreads, the movement of the rheostats will give a considerable flexibility in the lighting, even with the car in motion. The results of the initial settings and of the variations can then be studied with the car in motion at different speeds, with reference both to the conditions for which it was especially set and to the possible modifications suggested by driving over roads or under conditions of another type. After several such trials, one soon learns to build up beams that will meet a range of requirements and to form judgments as to the weight to be given to the various conditions.

The ammeter connections are arranged so that they will allow the reading of the current through one lamp while the others are burning and secure a record of a given set-up. A photometric test with the units set for the several values of current employed gives a quantitative record of the pattern and the intensity of the light.

The equipment, moreover, has considerable general utility. Thus, one can quickly mount pairs of head-lamps of any two types, for comparison under actual driving-conditions. The satisfactoriness of a given lighting-system is relative and, unfortunately, the impression left does not last long. Only by the immediate and direct comparison obtained by switching from one equipment to another can the true relative value in different respects be determined. The committee hopes that the facilities it has provided will find such general use that they will aid materially in the solution of the ever-present headlighting problem.

ACKNOWLEDGMENT

The Committee wishes to thank the several manufacturers for their cooperation in making the special parts of the testing equipments available. The task was not small, and the manufacturers who worked on the various parts put themselves to considerable inconvenience to make the test-cars available in time for this meeting.

METHOD OF OBTAINING VARIOUS TYPES OF BEAM PATTERN

BY J. H. HUNT[6]

ABSTRACT

SUPPLEMENTING Mr. Falge's description of the tests with headlighting devices, the author describes types of beam pattern for which the lamps were satisfactory and sketches a sort of contour map formed by the beam patterns, on which the beams are superimposed upon a given area, each successive beam being of smaller area and of higher candlepower than the preceding one. Provision was made for varying the depth of the outer area of comparatively low candlepower and for leaving in or removing the high intensity as seemed best, for moving the high intensity from the top to the center, and, finally, for building up a narrower beam in which the range of candlepower might be 600, 2000 or 10,000 for the respective areas. Two cars were equipped with special types of passing light, while two others were used because of their convenience in comparing different types of lamp. These cars are also described.

SEVEN cars have been equipped with special headlighting devices, as Mr. Falge has stated. Of these, five may be called S.A.E. test-cars, for the reason that the lamps have been mounted in the manner described by Mr. Falge. I shall describe briefly the types of beam pattern for which they have been set up, as samples of the type of work that might be done with this equipment, and shall sketch a sort of contour-map of the beam patterns, similar to a contour-map used in surveying.

The outline shown in Fig. 1 might represent an area 30 deg. wide and 4 deg. deep having at least 500 cp. within the entire area. Superimposed on this area might be a narrower beam, 18 deg. wide and 3 deg. deep, within which the candlepower is 2000. Finally, we might have an area of 12 deg. wide and 2 deg. deep in which the candlepower might be more than 10,000. It is possible, of course, to draw in a larger number of contour-lines, but this diagram will give a general idea of one type of beam that is to be demonstrated. Two cars have been equipped to work in this manner, a car supplied by the Nela Park engineering department of the General Electric Co. and a car supplied by the Guide Motor Lamp Co.

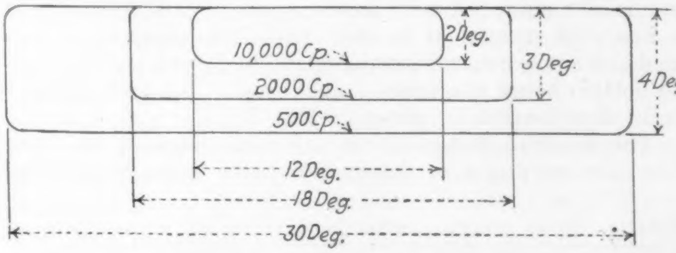


FIG. 1—CONTOUR MAP OF HEADLIGHTING
Over an Area Approximately 4 x 30 Deg. in Size at Least 500 Cp. Is Distributed. Superimposed on This Area Is a Beam, 3 x 18 Deg., Having a Candlepower of 2000. A Third Beam of 10,000 Cp. Is Then Spread over an Area of 2 x 12 Deg. Other Contour Lines May Be Drawn in Similar Manner

On these cars provision has been made for varying the depth of the outer area of comparatively low candlepower and for leaving in or removing the high intensity, as seems best; for moving the high intensity from the top to the center; and finally, for building up a narrower beam in which the range of candlepower might be 600,

[6] M.S.A.E.—Head of the electrical division, General Motors Corporation Research Laboratories, Detroit.

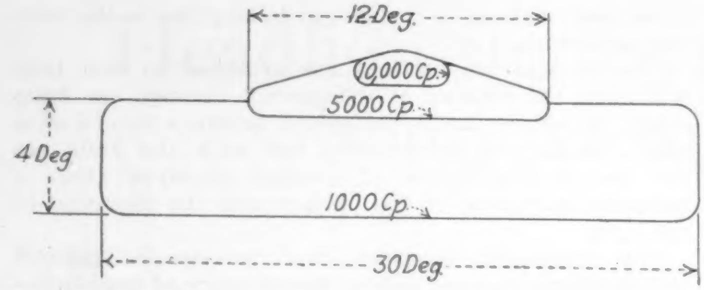


FIG. 2—ROUGH APPROXIMATION OF BEAM ON CAR SUBMITTED BY DR. H. C. DICKINSON

The Beam Pattern Is Triangular and Has Provision for a Relatively High Intensity at the Top of the Beam. The High-Intensity Portion of the Beam Can Be Cut Out, When Passing Other Cars, the Effect Being That of a Depressed-Beam System

2000 or 10,000, for the respective areas. In these latter beams, the candlepower is varied in a vertical direction only and the horizontal variation is kept to the minimum.

TRIANGULAR-BEAM PATTERN

Dr. H. C. Dickinson, of the Bureau of Standards, has cooperated in bringing a car that has been equipped with the triangular-beam pattern, having provision for a relatively high intensity at the top of the beam, as shown in Fig. 2. The high-intensity portion of the beam can be cut out when passing other cars, the effect being that of a depressed-beam system.

The General Motors Corporation has sent one car, a Cadillac, that is equipped with a beam pattern having an area approximately 5 x 36 deg. Provision has been made for getting the high intensity more nearly at the top of the beam pattern than would ordinarily occur with the simple spreading lens, by placing the filament below the reflector focus, which can easily be done with the equipment as provided. By this means, the beam is deepened and the intensity is thrown toward the top. The car is also equipped with another pair of lamps that have a beam 18 instead of 36 deg. wide, the same light flux and the same general arrangement of intensity from top to bottom being available in each case. The approximate light distribution is shown in Fig. 3.

The General Motors Corporation has another car that has been arranged to demonstrate two types of passing light. One type has a 40-deg. beam that has been deepened somewhat by putting one of the narrower spreading lenses crosswise, that is, one of the lenses that Mr. Falge has described as providing vertical distribution. This is to illustrate the possible advantage of a very wide beam on the down light. The width of the road that can be covered when the beam is depressed decreases for a given angle of light dispersion and these lamps show the advantage of spreading out the beam.

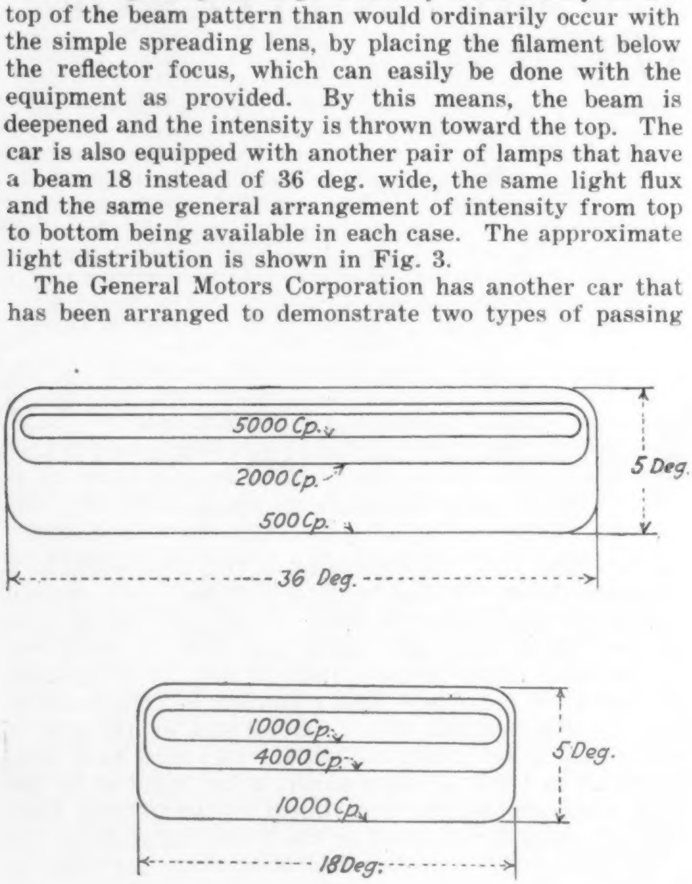


FIG. 3—BEAM PATTERNS OF A CADILLAC CAR EQUIPPED BY THE GENERAL MOTORS CORPORATION

By Placing the Filament below the Reflector Focus, the Beam Is Deepened and the Intensity Is Thrown toward the Top. The General Arrangement Is the Same in Both Illustrations, the Upper One Being 36 Deg., the Lower, 18 Deg. Wide

The uniform light intensity in the beam gives an impression of more light immediately in front of the car than farther away. The beam pattern is fairly uniform. On this car the other two lamps have been arranged with a 32-deg. spread of fairly good depth, the right-hand lamp being rotated counter-clockwise and the corner of the beam being raised, giving the effect shown in Fig. 4, the idea being that the rays from the upper right-hand corner of the beam will reach farther and will illuminate the ditches to a greater degree than those of a beam having a uniform 2-deg. depression below the horizontal.

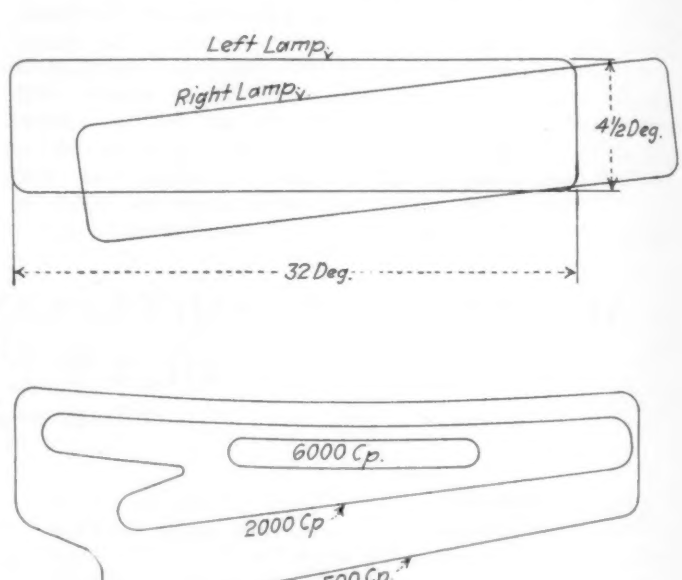


FIG. 4—TWO TYPES OF PASSING LIGHT

In the Upper View, the General Arrangement Is Shown; in the Lower, the Beams Are Superimposed, with the Left Lamp Bright. When the Right-Hand Lamp Is Rotated Counter-Clockwise, the Corner of the Beam Is Raised, the Idea Being That the Rays from the Upper Right-Hand Corner Will Reach Farther and Will Illuminate the Ditches to a Greater Degree Than Those of a Beam Having a Uniform 2-Deg. Depression below the Horizontal

Finally, we have two cars that have been used for testing-work by the General Motors Corporation that were equipped before the present S.A.E. equipment was available. These outfits have been presented to this meeting, not with the idea of attempting to sell this particular system, but with the idea of showing a type of test car that is very convenient to use in comparing different types of lamp.

COMPARISON OF BEAMS BY SWITCHING

I should like to endorse what Mr. Falge has said about the great advantage of being able to make a practically instantaneous switch from one type of equipment to another. These particular cars have been used to compare different types of depressed beam with the Crane system or with different types of depressed beam or different types of lamp arranged in accordance with the Crane

system. The car has four head-lamps, only two of which can be used at the same time. In one pair of parabolic reflectors, the upper third of the reflecting area is changed to the elliptical shape. That consequently throws light down and gives a deeper beam, even when the filament is in focus. It is equipped with a simple spreading lens, giving about 32-deg. spread. The light distribution is very close to that of the wide beam shown in Fig. 3.

The intensity is kept fairly well up to the top. The right-hand lamp has been rotated about 10 deg. in a counter-clockwise direction, when the beam pattern is viewed from the driver's seat. The superposed beam-patterns are shown in the sketch reproduced in Fig. 4. This is the arrangement described by Mr. Crane in his paper[7] read before the Detroit Section in November, 1925.

The switching is arranged so that, in passing other cars, the left lamp only is dimmed. This left lamp is deliberately pointed higher than the S.A.E. requirements, to get a good driving-light, and the passing is taken care of by dimming, as has been mentioned before. About the center line of the car there is approximately 1½ or 2 deg. of depression of the light from the right-hand lamp so that, when another car is encountered while only the right lamp is bright, the driver is not subjected to the glare to which he would be if the beam were thrown directly into his face.

Eliminating Lack of Symmetry

The other pair of lamps on this car are special reflector lamps that have been developed with the idea of eliminating the lack of symmetry in the total beam that comes from the method of superimposing two rectangular beams, which gives the appearance of a rather bright spot at the left and in front of the car. To do so, a beam pattern has been worked into a form like that shown in Fig. 5 and having a rather hot spot that is available in the center. The right lamp lies crosswise as shown, the two fitting together pretty well. This hot spot in the beam from the right lamp lights the road at a point near that toward which many persons point the spotlight. The seventh car has only one pair of lamps, the arrangement and beam pattern being like that shown in Fig. 5.

I should like to amplify one point already mentioned by Mr. Falge. In the past, types of lighting on motor-cars have been determined largely by what the lamp manufacturer has submitted to the car builder, and the only method that the car builder has had of deciding what type of lighting he wished was to make tests on such lamps as were available. The decision frequently was made by the purchasing agent as well as by the engineering department; but some type of lighting was

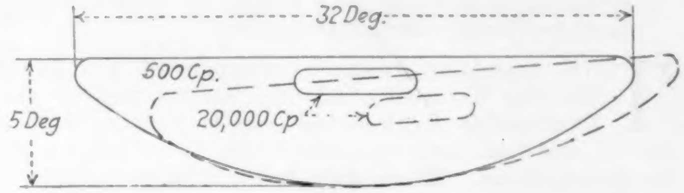


FIG. 5—ARRANGEMENT OF BEAM PATTERN FROM SPECIAL REFLECTORS
These Are Intended to Overcome the Lack of Symmetry in the Total Beam Produced by Superimposing Two Rectangular Beams, Which Gives the Appearance of a Bright Spot at the Left and in Front of the Car

arrived at. The lighting question has become so important that it is one of the questions most vitally affecting the relation of the car builder to the public; and we cannot continue to proceed on this basis. I do not wish to have what I am saying interpreted as a criticism of the work of the lamp manufacturers. Lamp manufacturers have been just as sincere in their belief that they have been giving to the public what the public wants as the car builders have been in supplying a given type of automobile built in just the way they have decided to make it. This responsibility is now becoming so great that the manufacturer cannot pass it on. We have reached the point at which the manufacturer really must specify the beam and, to do that, he must study the beam.

Importance of Study by Responsible Observers

I have found, in the course of something like 7000 miles of driving at night in the last 6 months, that my first impressions on many things were wrong. If each car builder equipped a car with the apparatus described by Mr. Falge and saw to it that this car was driven by responsible men from his engineering department, it would probably cause a revision of opinion as to the requirements of good lighting and would surely educate the manufacturer to the importance of studying the question and understanding it, the result being of great benefit to himself and to the industry at large. It is to this end that the committee has been working. As Mr. Falge has pointed out, to get almost any desired type of beam pattern from the equipment laid out is comparatively easy. Criticism has been raised that it is not possible to get some types of beam pattern easily. I will admit that; I think that the other members of the committee are willing to admit it. If anybody has a specific beam-pattern that he feels cannot be obtained with this type of equipment and will tell the committee what he wants, we shall be very glad to work on it and see whether we cannot find a way to produce it.

CAUSES OF DEFECTIVE HEAD-LAMPS

BY A. W. DEVINE[8]

ABSTRACT

A SURVEY is being made that is intended to bring out the defects in the design and construction of head-lamps and to show the causes of failure to secure in service the results for which the equipment was designed. Inferior head-lamp equipment supplied with the automobile and its improper use by the car owner or operator have thus far prevented the best results from being obtained. Headlighting should be handled as a safety matter without restrictions being placed upon the engineer in charge of the work. Suitable specifications should be written for the construction of the head-lamps and manufacturers should be held to strict compliance with the specifications. Car builders should cooperate with the State administrator and do all in their power to make the highways safe. The most common causes of trouble in head-lamps are improper mountings of the lamp and those resulting in wide variations from the designed optical characteristics; these are usually produced by inaccurate reflectors and inaccurate positioning of the light source. Lamp equipment should be built so that proper results will be obtained when the equipment is used by the average motorist. Simplicity is especially important. Adjust-

[7] See THE JOURNAL, May 1926, p. 467.

[8] M.S.A.E.—Engineer in charge of equipment section, registry of motor vehicles, Commonwealth of Massachusetts, Boston.

ing mechanisms should be as simple as possible and should be placed in accessible positions.

THE head-lamp research program of the Joint Steering Committee of this Society and of the Illuminating Engineering Society includes a survey of new head-lamp equipment that I am making at the present time. This survey is intended to bring out the defects in the design and construction of head-lamps and to show the causes of failure to secure, in service, the results for which the equipment was designed. My paper is based on a collection of part of this data.

The fact that considerable study is now being given to the subject and the possibility that a better system may yet be devised do not absolve us from the duty of perfecting the present system. Such perfection will give a practical demonstration of its value.

The difficulties that have thus far prevented the best results being obtained from this system are inferior head-lamp equipment supplied with the automobile and improper use of the equipment by the car-owner or operator. Equipment of durable construction, as far as possible free from complication and with optical characteristics that are definitely good, when such equipment is properly used, will bring the car-owner to a quicker realization of what is desired and will make him feel entirely dissatisfied with any but the proper results. An interest in his headlighting equipment will thus be fostered and enforcement will be made practicable.

Improvement of Equipment a Safety Matter

The improvement of new car-equipment has seemed to me to have been seriously handicapped, in general, by a lack of proper understanding of the problem by the men who determine the policies of the automobile-building companies. It should be given every consideration and handled as a safety matter, without the imposition of unfairly restrictive company policies upon the engineer in charge of that branch of the work. If this matter is properly handled, it will obviate the necessity for more restrictive regulation by State authorities. I refer to the car builder because he is the one who is responsible for the headlighting equipment on the cars that he makes and sells for the use of the motoring public. He should write suitable specifications for the construction of his head-lamps and should hold the lamp manufacturer to compliance with those specifications.

Any criticism of head-lamps on the grounds that they are not in compliance with the designed sample really goes back to the car builder who is not holding the lamp manufacturer to construction specifications. In the headlighting problem, the car builder should cooperate with the State administrator and do all in his power to make the highways safe.

Attention to Details of Construction

The attention of the automotive engineer should be given to every detail of head-lamp construction, lamp doors, door-attaching mechanism, front-glass-retaining mechanism, reflector accuracy, method of holding reflectors in the lamps, socket-sleeves, incandescent lamps, wiring, the focusing-mechanism, lamp housings, connectors, mountings, and mounting-supports.

I will cite one case that illustrates the difficulties that we encounter. One head-lamp equipped with a reflector of parabolic design was adjusted and tested, and the shape of the reflector was found to be such that its front and rear zones, when separated by interposing a 4-in. circular blank, or a card with a 4-in. circular hole at the reflector opening, had two different focal points $\frac{1}{8}$ in. apart on the reflector axis. Needless to explain, in a reflector of such design, the different zones should have a common focal-point.

Some Drivers Glared by Their Own Head-Lamps

If this lamp were mounted 36 in. high on a car, focused as well as possible and aimed, with due allowance for the loading of the car, the center of the main beam and probably the point of maximum intensity of the beam would strike the road at a point about 25 ft. in front of the car. This is a common reason for dissatisfaction with results in service and indicates the necessity for better lamp-construction. It should be obvious that a lamp which projects its beam center, and presumably its point of maximum intensity, to a point on the road surface 25 ft. in front of the car is unsatisfactory in practice. If the maximum intensity were very high, the driver would be glared by his own headlights; that is, the contrast between the illumination of the road at the point of maximum intensity referred to and at other points beyond would make it impossible for him to see objects at the usual distances of from 150 to 200 ft. required by State laws.

The operator of a car so equipped and adjusted would complain that he had insufficient light for safe driving and would criticize the present system of headlighting control. He would be likely to tilt the lamps up 6 or 7 deg. to bring the point of maximum intensity near the horizontal and eliminate contrasts, thereby improving his own driving light. It should be noted, however, that he would thus be partly correcting the inaccurate construction of his equipment at the expense of other motorists, measured in glare.

The details of head-lamp construction that are the most common causes of trouble are improper lamp mountings and those resulting in wide variations from designed optical characteristics. I am stressing these two points at this time because they demand immediate attention. The survey that we are making clearly indicates a wide variation in beam characteristics from lamps that should give similar results and also a number of cases of insecure lamp-mounting arrangements.

The variation from designed optical characteristics is usually due to inaccurate reflectors and inaccurate positioning of the light source. The filaments may be off center in the bulb, or the whole incandescent lamp may be out of position in the reflector; that is, the filament is not in its proper position relative to the axis of the reflector.

Suggested Improvements

What can the manufacturer do to improve this condition? First, he should build his equipment, or have it built, so that proper results will be obtained. Secondly, he should make an effort to get the point of view of the motor-vehicle administrator. The motor-vehicle administrator has to deal with persons whom I may call the average public, and they may be rich or poor, intelligent or unintelligent, careless or careful, reckless or cautious; they may be criminals or law-abiding citizens. It is the duty of the car builder to put lamps on his cars that the average motorist can use and from which satisfactory results may be obtained. Any complication in construction that makes the assembling of the head-lamp difficult for the average motorist should be given careful consideration. I wish to stress particularly the necessity for simplicity in the provision for head-lamp adjustment. Adjusting mechanisms should be as few and as simple as possible, and they should be placed in accessible positions.

STUDYING THE FIRST PRINCIPLES OF HEADLIGHTING

BY H. M. CRANE[9]

ABSTRACT

THE purpose of the testing devices that have been described is to enable everyone to get back to first principles in experimenting with headlighting equipment. Heretofore no way has been available to find out whether the undesirable features of a lamp could be modified without sacrificing its good qualities. Installations on the two General Motors cars have been arranged to (*a*) indicate the difference in driving with a very narrow and with a very wide driving-beam and (*b*) show the passing light that is obtained by depressing the total width of the beam in one case and depressing only the left side of the beam in the other. Investigation should be made of means of modifying existing equipment before the lamp manufacturers are asked to build new equipment to meet new requirements.

I WAS very anxious to have these test-cars brought here, to give everyone a chance to get back to first principles in experimenting. In the past we have invariably been asked to test complete outfits; that is, a pair of head-lamps has been supplied by a company, saying "This is the finest thing in the world; it does everything." Many times it does some things very well; but no way is available of modifying the undesirable features that the lamp may seem to have, to find out whether it can be modified without sacrificing its good qualities; and we generally end up by making the best guess possible and going ahead on that basis.

We have had some very good headlights in spite of this fact. But Mr. Devine has told you very clearly that we are badly in arrears in our duty to the public in supplying real lighting after dark. Not only can we make what we supply better, but we can also design it considerably better.

The diagrams that Mr. Hunt has shown were from head-lamps installed on two General Motors cars at my suggestion. One installation is arranged to give an indication of the difference in driving with a very narrow driving-beam and with a wide driving-beam; just two qualities in the lighting, without any frills and without particular consideration being given to passing light; just a chance to test one distinct quality of the headlight.

Arrangement of Passing Light

The other is arranged simply to show the passing light that is obtained by depressing the total width of the beam in one case and that obtained by depressing only the left side of the beam in the other. That, again, is reduced to as simple a thing as we could make on one car. I feel very strongly that we must go back at least as far as this, and I hope we shall go back even farther in investigating the subject before we begin asking the lamp companies to build new equipment to meet new requirements that I am sure will come from a study of the equipment that has been described.

I think the best thing to do now is to see what the equipment is like. I am sure the Committee will be glad to receive suggestions as to how the equipment should be modified. I made a few suggestions before I saw the equipment and others could easily make many more. As Mr. Hunt has said, if there is anything in the way of a beam that apparently cannot be produced by this equipment and that might be desirable to have on a car, suggest it and a way will be found to put it on a car for testing.

BEST USE OF LIGHT DEPENDS ON PERSONAL JUDGMENT

BY DR. C. H. SHARP[10]

ABSTRACT

AFTER equipment has been found satisfactory experimentally, its value must be determined by use. The answer will be largely a question of personal judgment as to what use should be made of the limited amount of luminous flux at our disposal so that it will be most serviceable to the driver. The answer will be found in the consensus of opinion of a large number of competent observers after they have formed fixed opinions of their own individually. When the question has been settled, it will probably remain so until someone appears with a radical improvement.

IT seems to me that those who have in charge the matter of the equipment and its operation for experimental work have it in good shape; the next thing to do is to use it. To get the answer from it will not be a simple matter. When all has been said and done, a very large element of personal judgment enters into these matters; perhaps we might call it a question of personal taste. We know that it is not humanly possible to get as much light in front of a moving motor-car as we should like. We know that, with the very best illumination that we can have, there are objects on the road at night that we ought to see and cannot hope to see. A black object on a black road does not show up with any amount of light that can be put on it. On the other hand, a white object against a black road will show up with very little light.

The question of judgment then enters as to just where, within practical limits, the line should be drawn; what use shall be made of the limited amount of luminous flux at our disposal, to make it most useful to the driver. This is a question, in my judgment, that can be answered only through practical tests on the road, made with an equipment that can be adjusted to meet various ideas as to what that best judgment can be. The test equipment described meets those requirements. The answer will come in the consensus of opinion of a large number of competent observers after they have given the question

[9] M.S.A.E.—Technical assistant to the president, General Motors Corporation, New York City.

[10] M.S.A.E.—Technical director, Electrical Testing Laboratories, New York City.

a sufficient amount of time and consideration to get fixed opinions of their own, individually. This is not a small or an easy matter to undertake, but the object is worth the effort. If the answer can be found, it will be an answer that will stand until someone appears with a radical improvement.

Complementary-Color Headlighting

We have heard a very interesting attempt to solve the problem in another way[11]. We are not prepared, of course, to say offhand whether Mr. Chambers has found a solution, because a great many practical questions will arise. No matter how good his physics may be, his plan may or may not succeed as an engineering solution. We always hope for improvements. Until the improvement comes, as has already been said, it is the duty of everyone concerned to make the best of the present system; and that is what the Steering Committee is trying to do. Mr. Hunt and Mr. Falge have indicated that very encouraging progress has been made in the work.

THE DISCUSSION

L. C. Porter[12]:—We have been interested in the problem from a somewhat different point of view; rather than finding out what is the best pattern of light on the road, we have been interested in finding out what happens when lamps get out of adjustment, when the filaments in the incandescent electric light sags a little bit or something else of that sort happens.

At the last Annual Meeting I showed some curves and photographs[13] that indicated the effect produced. Now we have carried that work a little farther and, at the Edison Lamp Works, have built a test-car that differs somewhat from those of the General Motors Corporation in that each of the various head-lamps is equipped with micrometer focusing-adjustments so that the light source can be moved in any direction in absolutely accurate steps of 1/64 in. We hope that, by using the experimental car on the road, we shall be able to obtain data that will bear out the laboratory data which I reported in Detroit[13], that other data will be added to the desirable beam-pattern which we expect you will secure with your cars, and that one more step will be taken in the head-lighting industry.

Question:—Headlights can be improved, of course, but can hardly be made foolproof. Why not take the "slant" of the human element into consideration, give the driver good lights, including a spotlight, and make it so easy to turn off the blinding lights that he will readily do so? Why not furnish a spotlight, encourage the use of it, and get rid of the blinding light in regular headlights?

R. N. Falge:—The test-car equipment is so constructed that a system such as you propose could readily be arranged and submitted by one of the participants in the investigation. Your suggestion really comes under the heading of depressible-beam headlighting, which happens to be a very live subject at the present time.

Depressible-beam head-lamps furnish the usual long-range driving-light, to satisfy the more difficult conditions of night-driving. In passing over hills, when the car is loaded, and at various other times, when a long-range beam would ordinarily be lifted into the eyes of an approaching driver and cause dangerous glare, the driver merely switches to a second beam that is aimed somewhat lower and is designed to keep the blinding intensity of the light well below the approaching driver's eyes.

A spotlight provides another means of obtaining a similar result. The spotlight beam can be aimed so that it will illuminate the road for about 100 ft. in front of the car and the depressible-beam feature is obtained by dimming the headlights. The spotlight beam is rather narrow and, in practice, is usually directed to illuminate only the right side of the road. Auxiliary driving-lights are available, however, that are especially designed for this service. They are fixed in position near the head-lamps and spread the beam over the entire width of the road.

J. W. Saffold[14]:—I wish to bring to your attention the point that a spotlight on a wet pavement, especially asphalt, gives a glare by reflection from the pavement. I have had some experience in that respect in night driving and, in that one particular, a spotlight gives as much glare as a headlight.

Mr. Falge:—A spotlight has so high an intensity that glare might result from light reflected from the pavement, if the beam were directly in line with an approaching driver. When the spotlight beam is directed toward the right side of the road, however, much less difficulty from this cause would be encountered.

Question:—Why not have a distribution that is in effect a modification of a fairly rectangular distribution of fair depth and considerable spread but having a high intensity confined to the upper right, a medium intensity low down on the left and low values in the rest of the pattern? In brief, why not use a distribution that has a localized bright spot at the right and upper part of the pattern instead of in the center, as most head-lamps have?

J. H. Hunt:—If that beam is intended to be a driving-beam, it would be subject to the objection that it is not a balanced beam and would not take care of turns of the road in either direction; the lighting, in that respect, would be very deficient. It would be fairly adequate on a straight road, would illuminate obstructions, such as cars parked close to the roadway and persons walking on the roadway, and would serve fairly well so long as a reasonable volume of oncoming traffic illuminated the left side of the road. But if a person attempted to drive over a winding road in a storm with such a set of lamps, he would find them very unsatisfactory.

Commenting on Mr. Chambers' paper[11], in view of the fact that I have at other times discussed some of the engineering difficulties in connection with making direct application of Mr. Chambers' idea, in that there is a loss of light-flux that requires more candlepower in the lamps to start with, reacts back on the generator and the whole electrical system and is liable to increase the cost, I think it is only fair to say that it does offer a means of meeting a difficulty that, so far as I know, none of the direct systems, as illustrated, will meet. On a rolling road, over short choppy hills, to attempt to make time with a directed-beam system is very trying, because, as you swing over the tops of the little rises, the beam is thrown down to the bottom and you cannot see what is on the next hill; you cannot see over it. If you had a type of lamp that would give illumination above the horizontal without the complication of glare to an approaching driver, you would have an all-round system that is not possible with the directed-beam system.

Mr. Crane:—The use of a type of head-lamp equipment in which one or more specially bright areas in the

[11] See The Journal, June 1926, p. 613.

[12] M.S.A.E.—Engineer in charge of special developments, Edison Lamp Works of the General Electric Co., Harrison, N. J.

[13] See The Journal, February 1926, p. 222.

[14] A.S.A.E.—President and sales manager, Devices Development Co., Cleveland.

light distribution are worked out is really a question of simplicity. Although it is not particularly difficult to make head-lamps to produce light distribution of the kind mentioned, this usually can be accomplished only by very close adjustment in centering the filament, whereas, to make the bright spots really useful, an extremely accurate adjustment of the head-lamp on the car, both laterally and vertically, is necessary. In the hands of the public, to get accurate vertical adjustment is much easier than accurate lateral adjustment and, at the same time, however accurate the lateral adjustment may be, it is rendered valueless by bends in the road. I am very sure that, to be of value, specially bright spots must be controlled to an accuracy of 1 deg. or better. If this is not done, they will be off the road on one side or the other and thus will not accomplish the desired result. If we can keep the light adjustment approximately right, vertically, we shall be doing all that can be expected while the public is handling high-candlepower beams. I believe that tests will show that illumination of the order of 40,000 or 50,000 cp. is entirely unnecessary, and I certainly hope that it will prove to be unnecessary, because of the difficulty of controlling the direction of high-power beams on so unstable a base as a motor-car hung on springs and working over very irregular road surfaces, both vertically and horizontally.

CUSHION TIRES AND THEIR RELATION TO THE VEHICLE-OPERATOR'S TIRE PROBLEM

(Concluded from p. 110)

cushion tires decrease skidding and forward sliding when the brakes are applied forcibly and therefore reduce the number of accidents. A truck that is involved in an accident may be laid up for one or several days with a loss of $30 or $40 per day in addition to the cost of repairs, which may amount to several hundred dollars. Insurance premiums on a fleet insured on an experience basis are reduced as the number and seriousness of accidents are reduced. A survey on a fleet of 19 trucks operating in New York City, made by an independent engineer, revealed that since cushion tires were used the accident premiums were lowered $1,600 per year, or practically $80 per truck.

Greater daily mileage per truck can often be obtained by the use of cushion tires, which give better riding-qualities than solid tires, thereby permitting higher speeds, and occasion fewer delays than pneumatic tires. This lowers the yearly operating-costs per mile and also increases the number of unit products delivered per day.

The elimination of emergency tire changes dispenses with the need of repair crews and eliminates emergency service calls, releases the inventory investment in spare tires and avoids loss through the theft of spare tires.

Cushion tires suffer some failures, to be sure, but these usually are the result of improper selection of size, improper truck maintenance, overloading, or overspeeding. The improper selection of size is productive of an excessive rate of wear and damage and, in some cases, of devulcanization, commonly known as blow-out. Sometimes damage involving base separation occurs. Poor truck maintenance that does not promptly correct wheel misalignment, loose bearings and incorrect brake adjustment often results in rapid and uneven tire wear. Overloading and overspeeding cause base separations and blow-outs.

Suitable Tire Equipment for Different Operations

Types of tire equipment that are regarded as correct for different classes of work are shown in a general way in Table 2, by which it will be seen that there are relatively few classes of trucks or motorcoaches on which cushion tires are not regarded as suitable equipment. The high-speed heavy vehicle that is operated on long trips must be equipped for the present with pneumatic tires, as solid and cushion tires cannot withstand the high heat that is generated in them in such service. I expect, however, to see a reduction from the high speed-capacity of some of the trucks that are now being built and also to see an increase in the speed capacity of cushion tires, which will make it possible to use cushion tires in this class of service, with a consequent saving in operating costs.

TABLE 2—USUAL TYPES OF TIRE EQUIPMENT REGARDED AS CORRECT FOR DIFFERENT CLASSES OF WORK

Vehicle	Speed, M.P.H.	Tire Equipment
½-Ton Truck	20 to 30	Pneumatic
1-Ton Truck	15 to 30	Cushion or pneumatic
1-Ton Truck	Over 30	Pneumatic
2-Ton Truck	15 to 25	Cushion or solid
2-Ton Truck	20 to 30	Cushion or pneumatic
2-Ton Truck	Over 30	Pneumatic
3½-Ton Truck	12 to 20	Cushion or solid
5-Ton Truck	10 to 18	Cushion or solid
7½-Ton Truck	8 to 15	Front, cushion or solid; rear, solid
Tractor	....	Cushion or solid
Urban Single-Deck Motorcoach	20 to 30	Cushion or pneumatic
Urban Double-Deck Motorcoach	15 to 25	Cushion or solid
Interurban Motorcoach	20 to 45	Pneumatic

Common Tire Failures and Their Causes

Some common types of tire failures or damage and the causes that contribute to them are as follows:

(1) *Rapid Tread-Wear on Front Wheels.*—This is caused most frequently by misalignment of the wheels. If the tires run other than parallel by only ¼ in. the tire must scrape sidewise about 70 ft. in 1 mile of forward running

(2) *Uneven Tread-Wear.*—This is often caused by loose shackles or spindles and, in the case of rear tires, by faulty braking. It also accompanies overloading

(3) *Flat Spots.*—These are often caused by the wheel being out of balance and resulting in a synchronous action at some speeds that leads to the formation of flat places on the circumference of the tire. Excessive play in the front-axle steering-connections is another common cause

(4) *Burn-Outs.*—These are almost always caused by excessive speed with normal load or overload, although the failure may not appear until a long time after the run in which the damage occurred

(5) *Base Separation.*—This may be started by any of a number of causes. It is started frequently by a hard blow on a tire base that overhangs the wheel felloe-band excessively. Another cause that often is not considered is overloading, which results in blows on the tire on rough pavement that are hard enough to start a rupture which spreads rapidly

SAFETY PROTECTION FOR HOLLAND TUNNEL

THE Holland Vehicular Tunnel between New York and New Jersey is to have the most elaborate provision for safety of its users of any place in the world. The Tunnel Commission's engineers have worked out a system for protecting motorists and methods for safeguarding this protective system that surpass anything of the kind that has ever been designed.

The tunnel will be illuminated by lamps spaced every 20 ft., except at the entrances, where they are spaced every 10 ft. to help in counteracting daylight. Each lamp is placed in an enameled reflector, set into the wall, and screened with a special frosted glass panel. Every other lamp will be supplied with power originating in New York and the remainder with power from New Jersey, and in addition, each group will have three independent sources of power supply. Each light and each ventilating unit can thus be supplied with power from any one of six independent sources. Four power houses must be destroyed before power for the ventilating system and the lamps will fail. In addition, the traffic signal lights will be automatically fed from an emergency storage-battery in case all other power should fail.

Traffic Signals

Three kinds of traffic signals will direct the motorists using the tunnel, green for "go ahead," red for "stop" and a special "stop-engine" signal. These lights will be spaced 240 ft. apart. Traffic officers located every 480 ft. will patrol the tunnel and each will have a traffic light control station and a telephone. Normally, only the green lights will show and the traffic will flow along as rapidly as possible. Should trouble arise, such as a collision or a car-fire, the nearest officer will promptly press a button, and all traffic lights up to the point of accident will then show red, thereby stopping on-coming traffic. The remaining lights will stay green, permitting the traffic ahead to clear itself. A hose and a chemical fire extinguisher every 120 ft. and a sand box every 240 ft. provide means for promptly controlling most fires, and tests have proved that the ventilating equipment will remove the fumes without danger to the occupants of the tunnel.

If the situation is such that it cannot be cleared up locally, the officer presses another button and summons one of the fire-and-wrecking trucks that are to be maintained in constant readiness at both ends of the tunnel. This truck will come in against the moving traffic, and as soon as it starts, a special yellow signal is flashed along the tunnel, notifying the officers of its approach and enabling them to clear the road.

When traffic can start again, the officer in charge will press a third button and the red lights will change to green. If in the meantime, however, an officer back along the line has for any reason pushed his "stop" button, the "go-ahead" lights will light up to that point only, the remaining lights staying red until changed at the proper station. Thus, the clearing of traffic at one place will not start traffic where trouble still exists.

Central Control Board

A board carrying 926 pilot lights will inform a central operator located in the administration building at the New York end of everything going on within the tunnel and will enable him to take full charge of any situation at any time. All changes in the traffic lights made by the traffic officers will be registered in the control room, and if necessary, the central operator himself will be able to stop traffic at any desired point. When this happens, the traffic officers in the tunnel cannot reset the signals; this can only be done by the central operator.

The pilot-light board in the control room will show, in addition to the position of the traffic lights, definite signals indicating the status and speed of every piece of ventilating apparatus and the condition of every circuit serving the tunnel. If a motor fails, a fan breaks down, a circuit goes dead, or a short-circuit develops anywhere, a bell will ring and a bright light, indicating the location and character of the trouble, will appear. The operator can stop the bell, but the light will continue bright until the trouble is corrected. When everything is in good order, the pilot lights will burn dimly, so that a bright light instantly attracts attention and calls for an investigation.

Ventilating-System Control

The 83 ventilating fans, located in the four ventilating-shaft buildings serving the tunnels, will be under the control of this central operator. He will vary the air supply in accordance with conditions within the tubes, providing an excess of pure air at all times. The operator will have sensitive instruments available which will show definitely the exact proportion of impure air. This will enable him to make a positive test of the purity of the air and detect small quantities of foul gases before they reach such proportions that they would be noticed in the tunnel. Six independent circuits will provide the power for these fans and duplicate equipment will always be in readiness to take the place of any fan that goes out of commission. If in spite of all these precautions the fans should fail and the air in the tunnel become too heavily charged with exhaust vapors, the "stop-engine" signal will be flashed, thereby preventing further contamination of the air until the ventilating equipment is put into operation again. As an instance of the refinement of this system, the "stop-engine" signal cannot be displayed unless the red "stop" lights are first lit, thereby preventing careless drivers from colliding with those who have promptly stopped their engines.

Should anything go wrong in the central control room, the fans can be controlled independently in the shaft buildings where they are located. Normally the power supply for the fans and control equipment will be divided equally between New York and New Jersey, but any piece of apparatus can be supplied from any source the operator desires, and all switch gear is to be so interlocked that it is impossible to make a wrong connection.

Chromium—Its Properties and Uses

By Colin G. Fink[1] DETROIT SECTION PAPER

ABSTRACT

CHROMIUM is the most important element in the structure of the motor-vehicle, asserts the author, and if it were not for chromium steel we could not build our automobiles of today. Next to iron, it is the most important metal in our whole industrial art.

Chromium was first isolated as a metal by Vauquelin, a Frenchman, in 1797. It belongs to the same group in the old Russian periodic table as molybdenum, tungsten and uranium, and has an atomic weight of 52. When analyzed, part of it resembles silicon and part resembles magnesium. It is not a rare metal; the world production in 1926 was 150,000 tons, compared with 7000 tons of tungsten, 50,000 tons of nickel and 1,800,000 tons of copper. Chromite ore deposits are distributed widely in the earth but largely because of the cheapness of labor in Africa, about 70 per cent of our chromium metal comes from Rhodesia; some comes from India and some from New Caledonia. Various chromium minerals exist but the only one of commercial importance is chromite, which is a combination of iron, chromium and oxygen.

The distinctive properties possessed by chromium give it a wider application than that of any other metal. The more important of these characteristics are extreme hardness, exceptional resistance to high temperatures and corrosion and high reflectivity. Because it imparts toughness to other metals, about 45 per cent of the production is used in alloys. Because of the high melting-point of its oxide, about 35 per cent is used in refractories, such as chrome brick for lining open-hearth and blast furnaces. About 20 per cent of the production goes into chemical compounds. Its high reflectivity, which is not far below that of polished silver, its hardness and its resistance to corrosion make it superior to nickel and silver as a plating material. It can be deposited on all other metals, so far as is now known.

An alloy of chromium with nickel, iron and manganese, called nichrome, is used as an electric resistor in enameling and heat-treating; all low-temperature electric ovens use nichrome because it resists disintegration at high temperatures. Chromium has a melting-point of 2948 deg. fahr., compared with 2642 deg. for nickel and 2912 deg. for pure iron and pure cobalt.

Formerly a 2-per cent chromium steel was regarded as high in chromium. Then, by enriching the draft in furnaces, it became possible to produce 30-per cent ferrochromium, but it was difficult to reduce the carbon content to a low point until the Goldschmidt process and the electric furnace were developed. The greatest development of chromium metallurgy began with the introduction of electric ferrochromes. Sixty-per cent ferrochromes are now being produced commercially and command a price of 33 cents per lb. of chromium content because of their low carbon-content.

The Goldschmidt process of producing chromium metal, which was developed simultaneously with the electric-furnace method, consists in mixing green chromium-oxide with aluminum, which acts as a reducing agent. Hydrogen under pressure or the more active nascent hydrogen at atmospheric pressure will likewise reduce chromium oxide.

Stainless steel is made from 13-per cent chromium and iron but cannot be produced commercially in thin sheets unless the carbon content is kept down to 1 per cent. Owing to its remarkable resistance to oxidation at high temperatures and to its expansion coefficient, scale does not chip off of chromium on the second heating. Chromium plating does not peel off from the base metal. Due to its extreme hardness, it is easier to polish the base metal and then plate with chromium than to polish the deposited chromium. The uses to which chromium plating can be put are innumerable, as it is not attacked by salt water, uric acid, nitric acid, molten tin, and various other agents.

AT the Chemical Show in New York City a few years ago a chassis was displayed with a ribbon attached to each part and leading to a card that told what function electrochemistry performed in the production of the part. Over this chassis was a large sign reading, Niagara Falls Made Detroit Possible. The original large center of the American electrochemical industry was located at Niagara Falls. We, as electrochemists, look upon the automotive industry as one of the by-products of the electrochemical industry.

A close study of the metallurgy of the automobile parts reveals that one element is more predominant and important than any other; this element is chromium. If it were not for chrome steels we should not be able to build our cars. Chromium spells the success of the automobile today.

Chromium, the metal, is not very old; it is about the same age as the United States. It was first isolated in 1797, by Vauquelin, a Frenchman. He produced the metal from a Siberian mineral, a lead chromate, and presented his results before the French Academy of Science at Paris. He described the metal as very hard and having a brilliant luster and a very high fusing-point. He expressed the opinion, however, which proved erroneous, that chromium never would be very useful as such but only in compounds such as potassium chromate. No one had any conception at that time of the chromium-alloy work that was to follow.

Chromium Is Not a Rare Metal

Chromium belongs to a group in the old Russian periodic table in which chromium comes first, then molybdenum, then tungsten, and next uranium. Tungsten is also a very important member of the group so far as car construction is concerned. Chromium resembles other elements in the group; it is hard like the others, but in certain respects it differs. It has an atomic weight of 52, and when analyzed by the X-ray part of it resembles silicon and part resembles magnesium.

The question often is asked, Is chromium a rare element? I think it will surprise some to learn that more chromium than copper exists in the earth, and considerably more chromium than nickel. We see more nickel and copper, so gain the impression that chromium is rare but, as a matter of fact, rich deposits exist all over the world, whereas nickel is known to exist in only two deposits of any importance. When mineral deposits are plentiful the producers usually go where labor is cheapest, and labor is cheapest at present in Africa. Practically 70 per cent of our chromium comes from Rhodesia,

[1] Head, division of electrochemistry, Columbia University, New York City; director of research, Chromium Corporation of America, New York City.

some from India and some from New Caledonia. We could take care of our needs in the United States very well from the deposits in California and the West. but our labor charges are much higher than in Africa. Labor conditions here are such that crude copper from the Belgian Congo and other far-away places is refined in America although we have plenty of copper ore in this country.

Various chromium minerals exist but the only one of commercial importance is chromite, $FeOCr_2O_3$, which is a combination of iron, chromium and oxygen. Chemists who know of the strong oxidizing properties of chromates may wonder why we have ferrous oxide combined with chromic oxide. The stability at high temperatures is the important factor. Latter-day alchemists in Japan are transforming mercury into gold; perhaps some day, if we ever run short of chromium, we can get these new alchemists to mix silicon and magnesium and give us chromium $(Si=28)+(Mg=24)=(Cr=52)$.

As to relative output, the world production of chromium in 1926, in terms of metal, was 150,000 tons; of nickel, 50,000 tons; of copper, 1,800,000 tons, and of iron, 95,000,000 tons. So, chromium cannot be called a rare metal. The public is familiar with tungsten; a layman would not consider tungsten rare yet the total world's production of tungsten is only 6000 tons a year. With 6000 tons we take care of the whole tungsten-steel industry.

Diversity of Application of Chromium

When we analyze industrial figures we find that chromium is second in importance to iron. With the exception of iron, no metal is so important at this time as chromium in our whole industrial art. We could better dispense with a dozen other metals than with chromium. The reason for this importance of chromium in industry and in the arts is tied up closely with the distinctive properties of the metal. About 45 per cent of the total production of chromium goes into alloys, about 35 per cent into refractories and the rest, about 20 per cent, goes into chemical compounds such as bichromate of soda and the like. That is a remarkable condition. No similar category can be shown for iron or copper. Something distinctively different in chromium gives it this diversity of application.

One of the best bricks on the market today for lining open-hearth and blast furnaces is chrome brick, which is made from chromite ore and has the advantages of being neutral and of withstanding high temperatures. We have acid bricks, such as silica bricks, and basic bricks, such as magnesite bricks; the chrome bricks are neutral. They have a melting-point of 2200 deg. cent. (3992 deg. fahr.), which is approached only by some high-grade magnesite bricks which run 98 per cent pure and have a melting-point of 1950 deg. cent. (3542 deg. fahr.). Silica bricks melt at 1800 deg. cent. (3272 deg. fahr.), so the chrome brick has a higher melting-point than the silica brick. The maximum temperature in blast furnaces is about 1900 deg. cent. (3452 deg. fahr.), hence a brick that will stand 2200 deg. cent. allows considerable margin. Two years ago a tariff was put on magnesite, which was the best thing that could have happened for chrome brick. The tariff was lowered later but industry continued to use much chrome brick, having learned its advantage over magnesite brick.

Alloys Used in Automotive Industry

The largest consumption of chromium is in alloys. This is of very special interest to the automotive industry. We have different classes of alloys. The three most prominent ones are the chromium steels, such as chromium-iron-carbon, chromium-nickel-iron-carbon and chromium-tungsten-iron-carbon. The last alloy is the most important; a little vanadium usually is included in the alloy. Next in importance to the chromium steels we have the chromium irons, which include rustless iron. Finally there are the special alloys, such as Stellite, which is a chromium-cobalt-tungsten alloy. When Elwood Haynes made his first samples of Stellite in 1907 or 1908, some of the samples I received from him were chromium-cobalt-molybdenum, but practically all Stellite now is chromium-cobalt-tungsten. Molybdenum does not quite equal tungsten as an alloying metal, especially not when steel is used for high-speed tools. To this third important class of alloys belong also the alloys that are characterized by nichrome, which is a high nickel-chromium-iron-maganese alloy. This is of interest to the automotive industry because it serves as the electric resistor of enameling and heat-treating ovens. Practically all low-temperature electric ovens use nichrome.

The chromium steels are important on account of their hardness and toughness, which properties can always be traced directly to chromium. But chromium imparts resistance to corrosion, besides physical hardness and toughness. In the case of nichrome a fourth important property enters. Chromium resists disintegration at high temperatures. We know of no other cheap element that could be substituted which would give the same resistance to heat. Thus we have in the chromium alloys four highly valuable properties. What actually makes the chromium steel so hard is a question that has been studied for many years, not only here but in England and Germany.

Chromium Alloys Produced in Electric Furnace

Regarding method of production, it can readily be seen wherein electrochemistry participates. Formerly, 3-per cent chromium steel was regarded as a very high chromium steel. Then, by enriching the draft in oil or fuel-fired furnaces, it was possible to produce ferrochrome with as high as 30 per cent of chromium, but it was very difficult to keep the carbon-content down. Not until the alumino-thermic and the electric-furnace methods were developed could the carbon be kept low. If the hardness of chromium steels is due to the presence of carbides, it is necessary to start with a fairly low carbon-content. What is it that is remarkable about the new stainless steel? It contains 13 per cent of chromium, but we were able to produce 13-per cent chromium steel years ago. However, we could not readily keep the carbon down. Stainless steel cannot be made into knife blades unless the carbon can be reduced to 1 per cent. We could not make a commercial stainless steel until we had the electric furnace or had aluminum for the Goldschmidt process. So Niagara Falls helped Detroit.

The greatest commercial development in chromium metallurgy started with the introduction of low-carbon ferrochromes. Aluminum and the electric furnace enabled us to make not only 30-per cent ferrochromes but to turn out a 30-per cent ferrochrome that is very low in carbon. The electric furnace is of the closed type. The heat is generated electrically and can be regulated better than in a fuel-fired furnace. Furthermore, in electric furnaces it is not necessary to stop at 30-per cent ferrochrome; one of the products of the electric furnace is 60-per cent ferrochrome, which finds a big market.

The elimination of impurities from poor grades of ferrochromium means increased cost of ferro-steels, hence

ferrochromium made in electric furnaces is commanding a price of 33 cents per lb. of chromium-content as against the fuel-fired or high-carbon-alloy cost of 11 or 12 cents per lb. of chromium. The steel manufacturer is willing to pay the price for low-carbon alloys. The electric furnace can produce ferochrome all the way up to 98 per cent of chromium.

The other method of chromium metal production which was developed in parallel with the electric-furnace method is the so-called Goldschmidt process. This consists of mixing green chromium-oxide Cr_2O_3, with an excess of aluminum. After the reaction we have chromium metal plus aluminum oxide. This process is used very little today for the production of chromium or chromium alloys.

Of the various metals in the periodic group to which chromium belongs, molybdenum and tungsten are readily reduced by hydrogen. If an attempt is made to reduce chromium oxide with hydrogen, the furnace run for awhile and the product taken out, the green oxide still remains; it is not reduced by hydrogen under atmospheric pressure, but if the pressure is increased to about 5 atmospheres the chromium oxide will be reduced. If, instead of using hydrogen under pressure, the so-called nascent hydrogen is substituted, it is much more active than the ordinary molecular hydrogen.

Resists Oxidation at High Temperature

The properties that were prominent in the case of chromium alloys all come to our attention again when the metal itself is investigated. The metal, unless it is very pure, is extremely hard. It has remarkable resistance to oxidation at high temperatures. If copper wire is heated to bright red heat and allowed to cool, a red scale of copper oxide chips off because the coefficient of expansion of the scale is very nearly that of the metal underneath. A scale does not chip off in the case of chromium. The oxide that is formed at the first heating of chromium protects the metal on the second heating.

The melting-point of chromium is about the same as that of pure iron or of pure cobalt; it is decidedly higher than that of nickel, which is 1450 deg. cent. (2642 deg. fahr.) That of pure iron and pure cobalt is 1600 deg. cent. (2912 deg. fahr.) and that of chromium is about 1620 deg. cent. (2948 deg. fahr.).

Another interesting physical property of chromium is its reflectivity, or quality to reflect light. Freshly polished silver has a very high reflectivity and if silver is rated at 100 then chromium is 82. Unlike silver, chromium is not affected by sulphur fumes; it does not require constant polishing like silver. A mirror having a constant reflectivity has been sought after for years. The best mirrors today are chromium mirrors. They are not made of solid chromium but are merely surfaced with the metal.

Noncorrosiveness of a High Order

The chemical properties of the metal are as fascinating as the physical properties. Its resistance to atmospheric corrosion is most remarkable. Copper and iron discolor quickly, whereas chromium metal remains bright indefinitely. Chromium is also extremely resistant to corrosion by liquids. One of the most corrosive liquids I know of is molten tin, which, when at a very white heat, is more corrosive than any acid. Tin at 1400 deg. cent. (2552 deg. fahr.) will pass through silica brick and magnesite brick. When we desired a material that would resist this molten tin, we tried chromium and found it to be unaffected. We placed a large sample of chromium metal in a tin bath at 1400 deg. cent. (2552 deg. fahr.) and after 1-hr. heating it was found that the sample had neither gained nor lost in weight. Accordingly, for extreme conditions of this kind it is a wonderful material to use as a lining for tin, brass or zinc furnaces. Furthermore, a number of zinc-roasting furnaces are now equipped with rabble blades made of high-chromium alloys.

The coefficient of expansion of chromium is very nearly that of glass, that is, decidedly lower than that of copper, iron or nickel and more nearly that of platinum, in fact, but a trifle higher. When chromium is heated to a high temperature a film forms on the metal. To eliminate scaling, it is necessary that the coefficient of expansion of the scale be the same as that of the underlying metal. Not only is this condition fulfilled but a second important condition of a truly protective coating is likewise fulfilled; namely, that the melting-point of the oxide film formed is about as high as or higher than that of the underlying metal.

Advantages of Chromium-Plated Ware

With the many valuable properties of chromium in mind we have for years devoted considerable time to the study of plating with chromium. When chromium is electro-deposited a very hard bright surface is obtained. We found that it was much easier to do all the polishing on the base and then do the plating than it was to do the polishing after the plating, because of the hardness of the chromium plate. Chromium is a passive metal and on this account has many valuable applications. Chromium is not attacked by uric acid, hence chromium-plated bathroom-fixtures retain their brightness remarkably well as compared with nickel-plated fixtures. When the metal is properly applied it makes a good finish for fittings for yachts and other sea craft. Fittings exposed to salt air continuously for months have resisted corrosion extremely well, much better than copper, nickel or brass.

The base of ordinary tablewear, such as knives, forks and spoons, is an alloy called German, or nickel, silver, which is an alloy of copper, nickel and zinc. A drop of water in the bowl of a chromium-plated nickel-silver spoon rolls around like a drop of mercury in a glass spoon, the water actually does not touch the metal, does not wet it and therefore cannot corrode it.

Chromium can be plated on brass, steel, aluminum, magnesium, silver, and other metals. I cannot begin to tell the many different applications. Plated iron pipes, 20 ft. long, subjected to repeated high-temperature heats remain unaffected. Chromium-plated surfaces exposed to nitric acid are remarkably resistant to this strong acid in which silver, copper, brass, and nickel readily dissolve. Chromium-plated golf clubs can be kept bright easily; the life of printing plates can be extended a hundredfold, and hardware fittings, automobile trimmings, engine parts, jewelry, reflectors, and many other articles can be protected from corrosion and wear by its use.

THE DISCUSSION

Question:—In depositing chromium, is it necessary that the anodes be the same shape as an article that is being plated?

Colin G. Fink:—No, not necessarily.

Question:—Is it possible to plate chromium on aluminum or duralumin cylinder-bores? It has been suggested for aeronautic engines.

Dr. Fink:—Yes, it can be plated on aluminum and on

magnesium with good results. It is always well to know what the part is to be used for after it is plated. We have not yet found a metal that we could not plate.

QUESTION:—What is the cost of plating with chromium?

DR. FINK:—It is much less than the cost of silver plating; it is nearer that of nickel plating.

QUESTION:—Does not chromium plate chip from iron very easily?

DR. FINK:—The one characteristic distinction between nickel and chromium plate is that chromium does not chip off as easily.

QUESTION:—Is not the use of chromium to produce hard surfaces for resistance to abrasion the field for which chromium is best adapted?

DR. FINK:—That question is answered differently by different men depending upon the fields in which they are engaged. The three important facts are: first, the plate can be made very hard, harder than file steel; second, the plate can be made very resistant to corrosion; and third, the plate can be made very resistant to disintegration at high temperatures. Personally, I consider the resistance to corrosion the biggest field today.

QUESTION:—What is the normal thickness of chromium plate?

DR. FINK:—About the same as nickel plate.

QUESTION:—What thickness do you regard as best?

DR. FINK:—It depends on the purpose to which the article plated is to be put. The more severe the service, usually, the heavier the plate required.

QUESTION:—Can a chromium-plated sheet be formed or stamped without peeling or chipping?

DR. FINK:—Yes, that we have done time and again.

QUESTION:—Can chromium be plated on zinc-base metal, and what finish can be obtained?

DR. FINK:—Yes, and with proper precautions any desired finish can be obtained?

QUESTION:—Is there any metal or coating material for steam-boiler construction that will not rust out?

DR. FINK:—The best plan I know of for steam-boiler construction is to use electrolytic protection and the best boiler tubes that can be made from steel rather than have the boiler parts made of special alloys, which do not meet the mechanical specifications of high-pressure boilers. In the Kirkaldy electrolytic-protection system an iron anode is inserted into the boiler shell which latter is kept at a cathodic potential.

QUESTION:—Can the plate be used on small tools, as on 1/8-in. drills? What can you say about the application to plating broaches, plug-gages, crankshaft pins, and the like?

DR. FINK:—It works very well, especially in the case of drills.

FRONT-END-DRIVE SYMPOSIUM

(Concluded from p. 140)

solid-type mounting. It is a 1/2-in-pitch-chain layout having sprockets of 21, 42 and 18 teeth, for an eight-cylinder-in-line engine.

Fig. 8 presents a layout showing the camshaft and the generator on opposite sides of the center line of the engine. The high position of the generator should be noted since it is placed there purposely to provide ample chain-contact on the crankshaft sprocket. This is a 4/10-in.-pitch layout having 24, 48 and 20 teeth in the sprockets. The generator is attached to a special flange, with only two cap-screw holes for mounting; one pivots the generator and the other is slotted with a slot wide enough to adjust the chain to its maximum life, or about 5/8 in. at the center of the sprocket. The inset in Fig. 9 shows the method of assembling the generator in position with the sprocket attached. Endless chains are usually furnished and are guided or retained on the sprocket wheels by means of a solid center-link, which follows a groove cut in the center of the face of all the sprocket-teeth. Some method such as this is devised to simplify the assembly operation.

Regarding the assembling with our manual adjustment, shown in Fig. 9, sometimes it is found impossible to assemble the accessory unit in place last. In this case, with our manual adjustment, it is necessary to flange the camshaft sprocket, attaching it with cap-screws over a short pilot and placing it in position.

Fig. 10 shows a two-sprocket drive without adjustment. With the use of a chain of liberal width, it is sometimes found possible to drive without adjustment when there is no small driven sprocket involved, and the element of wear is thus greatly reduced. A careful analysis of conditions must be made in an attempt of this character. The chain must be free of whip or other elements that would accelerate wear. Fig. 10 shows a small engine; a 1/2-in.-pitch chain 1 1/4 in. wide is used, with 17 and 34 teeth in the sprockets.

In all engineering work there must be a compromise in the design of various parts, and nothing is truer in that respect than the situation as to silent-chain drives. The object of their use is silence, with which must go satisfactory life, and both are controlled largely by design. Hence, considerable care should be exercised in this direction. The successful drives are those which have been developed after careful consideration of the advantages to be gained and the pitfalls to be avoided.

THE DISCUSSION

E. W. TEMPLIN[10]:—In regard to the silent-chain drive, what data are available to assure us that such a drive is dependable as to strength?

F. M. HAWLEY:—Dependability all hinges on the conditions of the drive and the nature of the service. Answering the question broadly, last year something like 1,500,000 cars were equipped with chains. This year, probably 2,000,000 cars will be so equipped. So, the dependability is partly explained in that way.

T. O. DAY[11]:—If the life of the silent chain is from 30,000 to 50,000 miles, how does the life of the sprockets compare with that of the chain?

MR. HAWLEY:—It is seldom necessary to replace a set of sprockets in less time than the length of life of two chains. The sprocket wear is normally very light. The crank and the generator sprockets are of hardened steel and the cams are of cast iron.

[10] M.S.A.E.—Engineer, motorcoach division, Six Wheel Co., Philadelphia.

[11] M.S.A.E.—Master mechanic, Yellow Cab Co., Philadelphia.

Applicants *for* Membership

The applications for membership received between Nov. 15 and Dec. 15, 1926, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

ANDERSON, W. R., development engineer, Manhattan Rubber Mfg. Co., *Passaic, N. J.*

BARBER, D. M., service engineer, Wayne Tank & Pump Co., *Fort Wayne, Ind.*

BAUER, GEORGE C., draftsman, Curtiss Aeroplane & Motor Co., *Garden City, N. Y.*

BECKER, GEORGE E., assistant engineer, casualty and automotive department, Underwriters' Laboratories, *Chicago.*

BELAND, EUGENE, assistant chief tool designer, International Motor Co., *Plainfield, N. J.*

BUHL, MARVIN H., tool design detailer, Buick Motor Co., *Flint, Mich.*

BURKE, J. W., garage foreman, Shell Co. of California, *Long Beach, Cal.*

CRAMER, PARKER DRESSER, aviator, Cramer Flying Service, Cramer Flying Field, *Clarion, Pa.*

CROMPTON, KENNETH, G., foreman, Crompton Machine Co., *Lawrence, Mass.*

DOBBE, FRANK A., engine layout mechanic, motorcoach engineering department, Yellow Truck & Coach Co., *Pontiac, Mich.*

DUCKWORTH, R. G., supervising road mechanic, Armour & Co., *Charlotte, N. C.*

DURLACHER, CYRIL, automobile engineer, 6 Upper Street, Martin's Lane, *London, W. C. 2, England.*

EDDY, RAYMOND, manager, General Truck Service Co., *Chicago.*

ENGELHARDT, NICKOLAUS LOUIS, JR., student, Sheffield Scientific School, Yale University, *New Haven, Conn.*

FAWLEY, ALBERT E., metallurgist, Standard Steel & Bearings, Inc., *Plainville, Conn.*

FISCHER, EMILE, mechanical engineer, International Motor Co., *New York City.*

FISHER, ALFRED J., vice-president in charge of engineering, Fisher Body Corporation, *Detroit.*

GODFREY, GEORGE B., special motorcoach representative, Reo Motor Car Co. of New York, Inc., *New York City.*

GRACE, JAMES W., superintendent of equipment, California Highway Commission, *Willits, Cal.*

GREENEBAUM, CHARLES L., president, Metropolitan Distributors, Inc., *New York City.*

HEARD, C. W., research engineer, *Corona, N. Y.*

HENDRICKSON, THOMAS C., lubricating engineer, Pure Oil Co., *Minneapolis.*

HORNE, CALEB L., mechanical engineer, Texas Co., *New York City.*

HUSSONG, PAUL, service manager, Greer College of Automotive Engineering, *Chicago.*

JACOBSON, BEN B., production manager, Stinson Aeroplane Corporation, *Northville, Mich.*

JONES, WALLACE F., sales engineer, Standard Steel & Bearings, Inc., *Plainville, Conn.*

KALLINICH, CHARLES J., sales engineer, Charles D. Schmidt Corporation, *New York City.*

KAW, H., draftsman, engineering department, Cadillac Motor Car Co., *Detroit.*

KIMSEY, JOHN FORSYTHE, superintendent, Spicer Mfg. Corporation, *South Plainfield, N. J.*

LANDRY, GEORGE J., sales manager, Gairing Tool Co., *Detroit.*

LEBARRE, ROWLAND S., Detroit sales manager, Interstate Iron & Steel Co., *Chicago.*

LIDKEA, H. J., designer, Kelly-Springfield Truck & Bus Corporation, *Springfield, Ohio.*

LUNDGREN, A. ARTHUR, secretary, Burd High Compression Ring Co., *Rockford, Ill.*

MACFEE, ROBERT, secretary, National Standard Parts Association, *Detroit.*

MATTHEWS, W. C., superintendent of motor vehicles, Norfolk branch, Standard Oil Co., *Norfolk, Va.*

MCLAUGHLIN, W. J., production manager and industrial engineer, Spicer Mfg. Corporation, *South Plainfield, N. J.*

MOTT, C. B., district manager, Reeves Pulley Co., *Columbus, Ind.*

NAYLOR, WILLIAM C., chief engineer, Stinson Airplane Corporation, *Northville, Mich.*

NEILL, THOMAS, junior engineer, automotive laboratory, Bureau of Standards, *City of Washington.*

NORVIEL, HARRY E., engineer, Remy Electric Division of General Motors Corporation, *Anderson, Ind.*

PATTON, FRED C., assistant manager, Los Angeles Motor Bus Co., *Los Angeles.*

PEACOCK, T. H., owner and general manager, T. H. Peacock, *Calgary, Alberta, Canada.*

PLATZER, GEORGE E., sales engineer, General Electric Co., *Schenectady, N. Y.*

POLLARD, THOMAS ROYLE, lecturer in electrical engineering, *Canterbury College, Christchurch, New Zealand.*

PYLES, RUSSELL, principal aeronautic draftsman, National Advisory Committee for Aeronautics, *Langley Field, Va.*

REID, D. B., service manager, Automotive Gear Works, Inc., *Richmond, Ind.*

RIGDON, T. CAROL, mechanical design checker, Buick Motor Co., *Flint, Mich.*

SALTUS, R. SANFORD, JR., vice-president, Ludington Exhibition Co., *Philadelphia.*

SCHANK, HARRY E., assistant engineer, McCord Radiator & Mfg. Co., *Detroit.*

SCHMID, WILLIAM, production manager, H. A. D. Sales & Engineering Co., *Bronson, Mich.*

SHAW, FRANCIS, inspector, Air Corps, *New York City.*

SIMMONS, CHARLES R., service manager, Studebaker Corporation of America, *Cleveland.*

SMITHERS, V. L., president, V. L. Smithers, Inc., *Akron, Ohio.*

STANLEY, J. R., manager, Stanley-Bradt Co., Inc., *Alexandria, La.*

TRUBE, ROBERT L., sales engineer, North East Electric Co., *Rochester, N. Y.*

VAUGHAN, F. M., traffic manager, Coventry & District Co-operative Society, Ltd., *West Orchard, Coventry, England.*

VAN DER BECK, JOSEPH, technical oil salesman, Standard Oil Co. of New Jersey, *Newark, N. J.*

VAN DERMARK, FRED J., tool designer, Buick Motor Co., *Flint, Mich.*

WELLS, FRANK W., service representative, Chrysler Motor Corporation, *Detroit.*

WERNER, WILLIAM, manager of production methods, Horchwerke Actiengesellschaft, *Zwickau, Saxony, Germany.*

WEST, CARL ORRIN, assistant electrical engineer in charge of operation, Bureau of Power & Light, City of Los Angeles, *Los Angeles.*

WILDEMAN, A. G., sales and service, K. P. Products Co., 6 Beaver Street, *New York City.*

WILSON, DONALD K., transportation engineer, Utica Gas & Electric Co., *Utica, N. Y.*

WINDBERGER, OTHMAR, technical director and engineer, Oesterr, Daimler Motoren Actiengesellschaft, *Wiener-Neustadt, Austria.*

YOUNG, RAYMOND, assistant engineer, Ross Gear & Tool Co., *Lafayette, Ind.*

Applicants *Qualified*

The following applicants have qualified for admission to the Society between Nov. 10 and Dec. 10, 1926. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff) Affiliate; (S M) Service Member; (F M) Foreign Member.

ALTMAN, P. (J) professor in the department of aeronautics, University of Detroit, 630 East Jefferson Avenue, *Detroit*.

BARNETSON, WILLIAM W. (A) superintendent, American LaFrance Fire Engine Co., *Toronto, Ont., Canada*; (mail) 112 Laughton Avenue.

BEHN, HAROLD E. (A) tool engineer, Locomobile Co. of America, Inc., *Bridgeport, Conn.*; (mail) 173 Madison Terrace.

BELLANCA, GIUSEPPE M. (M) consulting aeronautical engineer, Wright Aeronautical Corporation, Paterson, N. J.; (mail) 91 Henry Street, *Passaic, N. J.*

BERNHARD, EDWARD W. (A) manager of automotive equipment division, L. H. Gilmer Co., Tacony, Philadelphia; (mail) 4835 Woodward Avenue, *Detroit*.

BERTHOLD, GEORGE H. E. (A) engineer, Rajah Co., Bloomfield, N. J.; (mail) 241 South Burnett Street, *East Orange, N. J.*

BOWLES, JAMES T. B. (M) petroleum technologist, Crown Central Petroleum Corporation, *Houston, Tex.*

BRUSCHI, RAMBALDO (F M) engineer, Officine di Villar Perosa, *Turin, Italy*; (mail) Via Nizza 148.

BUSHKIN, WILLIAM H. (J) field service representative, Chrysler Sales Corporation, 341 Massachusetts Avenue, *Detroit*.

CANDLER, D. W. (A) general manager, Holley Carburetor Co., *Detroit*; (mail) 2434 Longfellow Avenue.

CHAMPION, LIEUT. C. C., JR. (S M) bureau of aeronautics, Navy Department, *City of Washington*.

CONE, H. I. (M) vice-president and treasurer, Daniel Guggenheim Fund for the Promotion of Aeronautics, 598 Madison Avenue, *New York City*.

DURHAM, FRED S. (A) vice-president and treasurer, Bonney Forge & Tool Works, *Allentown, Pa.*

EVANS, O. G. (J) sales engineer, Bearing Service Co., Proprietary, Ltd., 92 Flinders Street, *Melbourne, Victoria, Australia*.

FAGAN, THOMAS Z. (A) sales manager, Scintilla Magneto Co., Inc., *Sidney, N. Y.*

GOSSELIN, JOSEPH A. (A) service manager, Leghorn Motors Co., Boston; (mail) 11 Boston Avenue, *West Medford, Mass.*

HOOPES, CHARLES R. (A) president, Hoopes Bros. & Darlington, Inc., *West Chester, Pa.*; (mail) P. O. Box No. 64.

HUBER, HENRY (J) new development department, General Motors Truck Co., *Pontiac, Mich.*

HUGHES, ADRIAN, JR. (M) superintendent of motorcoach transportation, United Railways & Electric Co., 1314 Continental Building, *Baltimore*.

HURLIN, HENRY B. (A) inspector and tester in maintenance department, Rolls-Royce of America, Inc., *Springfield, Mass.*; (mail) 598 Dickinson Street.

IRELAND, G. S. (A) president, Ireland Aircraft, Inc., *Garden City, N. Y.*; (mail) 127 Chestnut Street.

JOHNSTON, JOHN OGLE (J) assistant works manager, Thornycroft of Australia, Ltd., P. O. Box No. 2622, *Sydney, New South Wales, Australia*.

JONES, G. LESTER B. (M) mechanical engineer, Sperry Gyroscope Co., *Brooklyn, N. Y.*; (mail) 35 Orange Street.

JOYCE, TEMPLE N. (A) special representative, Curtiss Aeroplane & Motor Corporation, Garden City, N. Y.; (mail) Racquet Club, *City of Washington*.

KNOX, GARNER L. (M) engineer, Utility Trailer Mfg. Co., *Los Angeles*; (mail) P. O. Box No. 206, Arcade Station.

LEVI, GASTONE (J) mechanical engineer, Scripps Motor Co., *Detroit*; (mail) 1509 Glynn Court.

LEWIS, H. S. (A) vice-president, Parish Mfg. Corporation, *Reading, Pa.*

LEWTON, ROSS E. (M) mechanical engineer, research department, Studebaker Corporation of America, *South Bend, Ind.*

LINN, ELLERY C. (A) sales agent, Linn Mfg. Corporation, Morris, N. Y.; (mail) 47 Van Wagener Avenue, *Jersey City, N. J.*

MEAD, HOWARD C. (A) chief draftsman, Guide Motor Lamp Mfg. Co., Cleveland; (mail) 2035 Richland Avenue, *Lakewood, Ohio*.

MULLANEY, RICHARD L. (J) mechanical engineer, Goodlin Automotive Equipment Co., *South Bend, Ind.*; (mail) 219 East Marion Street.

NENNINGER, L. F. (M) engineer, Cincinnati Milling Machine Co., Cincinnati; (mail) 2754 Erie Avenue, *Hyde Park*.

NICHOLLS, ERNEST R. (A) 54 Cottage Street, *Meriden, Conn.*

NICHOLLS, RAYMOND E. (J) engineering draftsman, 54 Cottage Street, *Meriden, Conn.*

NOLLER, CHARLES W. (M) experimental assistant, International Motor Co., Long Island City, N. Y.; (mail) 41-67 Forley Street, *Elmhurst, N. Y.*

O'DONOHOE, CORNELIUS M. (M) chief tool designer, Lycoming Mfg. Co., *Williamsport, Pa.*; (mail) 524 Louisa Street.

PARSONS, MAURICE B. (M) wheel engineer, Motor Wheel Corporation, *Lansing, Mich.*; (mail) 1208 North Chestnut Street.

PETRE, A. PRESTON (M) experimental engineer, American Hammered Piston Ring Co., *Baltimore*; (mail) 829 University Parkway.

PHILLIPS, ERNEST B. (M) chief chemist, Sinclair Refining Co., New York City; (mail) Sinclair Refining Co., *East Chicago, Ind.*

RANDALL, HENRY D. (A) manager of sales, non-metallic gear department, General Electric Co., Schenectady, N. Y.; (mail) General Electric Co., River Works, *West Lynn, Mass.*

REINHARD, ROBERT N. (A) manager of transportation, Burr Creamery Corporation, 798 Towne Avenue, *Los Angeles*.

ROUSTON, WILLARD R. (A) automobile mechanics instructor, Placer Union High School, Auburn, Cal.; (mail) P. O. Box No. 121, *Loomis, Cal.*

RUHF, C. T. (M) assistant factory manager, Mack plant, International Motor Co., *Allentown, Pa.*; (mail) 1202 South 10th Street.

RYCE, LUCIUS C. (A) *Fitzwilliam Depot, N. H.*

SAWYER, DAVID P. (A) sales manager, Christensen Air Brake Co., 6513 Cedar Avenue, *Cleveland*; sales manager, Duroseal Corporation, Cleveland; (mail) Christensen Air Brake Co.

SMITH, WESLEY L. (A) pilot, Air Mail Service, Hadley Field, New Brunswick, N. J.; (mail) 223 Wayne Street, *Highland Park, N. J.*

TAYLOR, THOMAS D. (J) metallurgist, Bliss & Laughlin, Inc., *Harvey, Ill.*; (mail) 47 East 154th Street.

TILLBERG, C. E. (A) shop superintendent, Twin City Motor Bus Co., Inc., *St. Paul, Minn.*; (mail) 407 Aurora Avenue.

TITTERINGTON, MORRIS M. (M) vice-president and chief engineer, Pioneer Instrument Co., Inc., 754 Lexington Avenue, *Brooklyn, N. Y.*

TOCH, MAXIMILIAN (M) vice-president, Toch Bros., Inc.; vice-president, Standard Varnish Works, *New York City*; (mail) 50 Central Park West.

TROTH, E. A. (A) district manager, Richfield Oil Co., P. O. Box No. 315, *Stockton, Cal.*

TROXELL, GEORGE B. (M) research engineer, Bethlehem Steel Co., Bethlehem, Pa.; (mail) 185 Charles Street, *Easton, Pa.*

WATSON, THOMAS A. (M) designing engineer, University of California, *Los Angeles*; (mail) 3419 Marathon Street.

WEEKS, ERNEST GEORGE (A) automobile trouble shooter, Troy Motor Sales, *Los Angeles*; (mail) 1011½ West 103rd Street.

THE

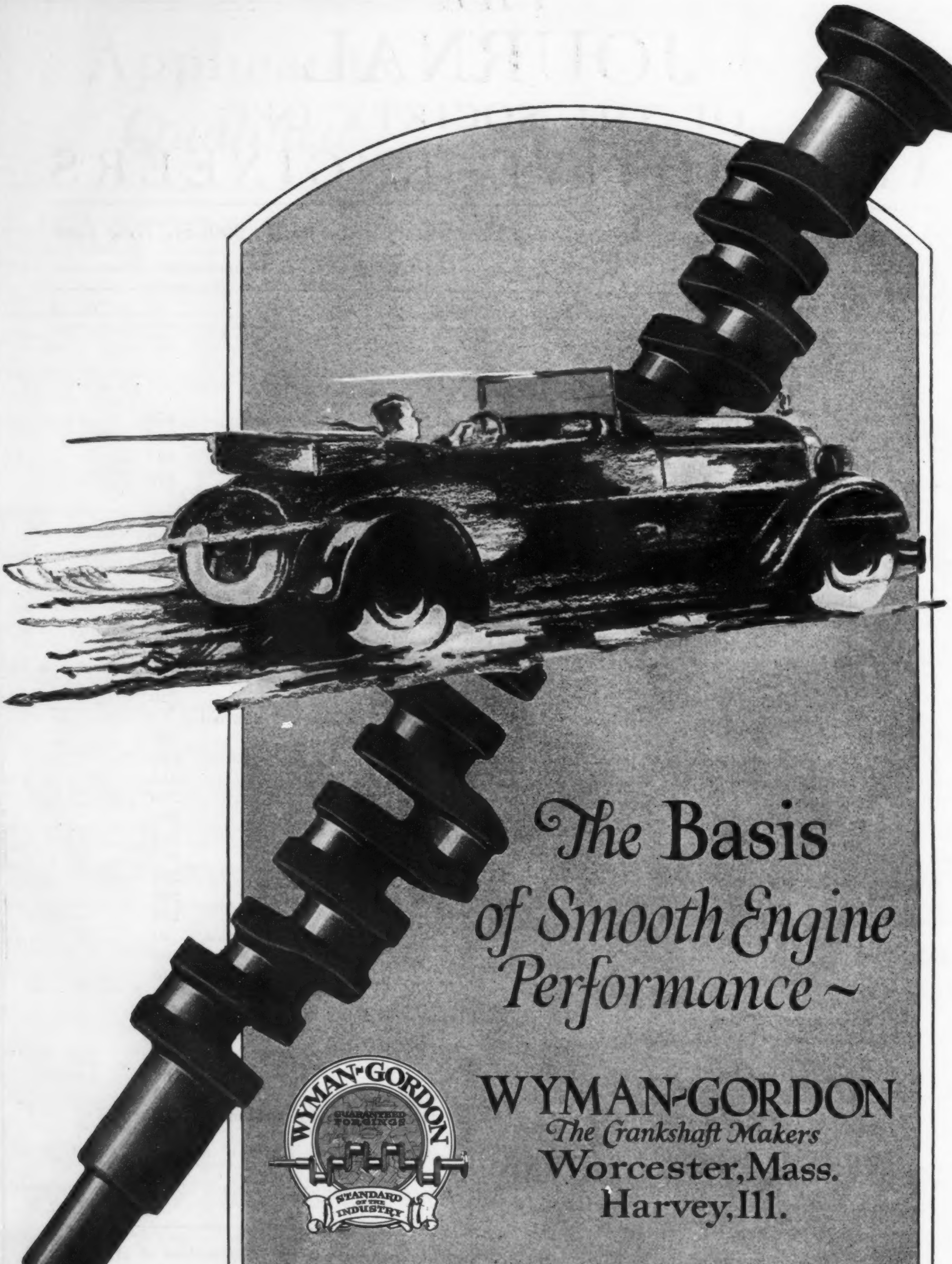
JOURNAL
OF THE SOCIETY OF
AUTOMOTIVE ENGINEERS

Published by the SOCIETY OF AUTOMOTIVE ENGINEERS, INC., *29 W. 39th St., New York*

J. H. HUNT, *President* COKER F. CLARKSON, *Secretary* C. B. WHITTELSEY, *Treasurer*

Vol. XX February, 1927 No. 2

INDEX


The Basis
of Smooth Engine
Performance ~
WYMAN-GORDON
GUARANTEED FORGINGS
STANDARD OF THE INDUSTRY
WYMAN-GORDON
The Crankshaft Makers
Worcester, Mass.
Harvey, Ill.